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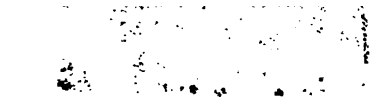
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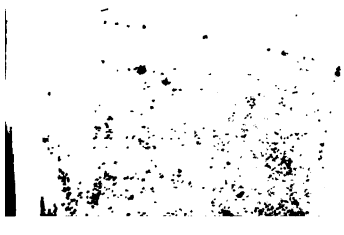
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BEARINGS
AND
THEIR LUBRICATION

Bearings

and

Their Lubrication

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1911
Published by the
American Machinist
McGraw-Hill Book Company, Sole Selling Agents
239 West Thirty-ninth Street, New York
London, E. C., 6 Bouverie Street Berlin, N. W. 7, Unter der Linden 71

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American Machinist, New York, U. S. A.

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PREFACE

The aim of this book has been two fold; to present the underlying principles involved in the design of all classes of machinery bearings and to show modern practice in the construction and application of important commercial types. It is believed to be the first treatment of these subjects ever put into book form.

The tendency of modern machine design in many fields is toward the use of high speed shafts and spindles and friction reducing forms of bearings. Therefore the present seems a peculiarly fitting time to present these data with the purpose of making them permanently useful to engineers, designers, draftsmen and machinists; in fact to any one who is interested in any way with machinery bearings and their proper lubrication and care.

Many of the data have never before been published in any form. Many of the experiments and researches quoted are from European sources and official transactions of technical societies and are not generally known to American readers.

It is a pleasure to acknowledge the many courtesies received from engineers and machinery manufacturers and the valuable information furnished by them. Without this hearty coöperation it would have been impossible to have produced this work.

L. P. ALFORD.

NEW YORK CITY.
August, 1911.

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BEARINGS AND THEIR LUBRICATION

CLASSIFICATION OF MACHINERY BEARINGS

Machinery bearings are the parts of the bed, frame or other members that constrain rotating parts, such as shafts and spindles. They are divided into two general classes, journal and thrust, depending upon the direction in which the load acts. In the journal bearing the load acts at right angles to the axis; in thrust bearings, parallel to the axis.

The bearing surface of all journal bearings is necessarily circular in cross-section. In profile it is ordinarily cylindrical, but may be conical, spherical or even of a more complicated shape. The bearing surface of thrust bearings is ordinarily flat, but may be spherical, conical, or shaped to the curve of the tractrix.

Bearings are also divided into two classes by the kind of contact between the surfaces; that is, bearings having sliding contact, or ordinary bearings, and bearings having rolling contact or ball and roller bearings.

The bearing surfaces of ball bearings both radial (journal) and thrust are ordinarily curved races with a circular cross-section. Less commonly the surfaces are flat.

The bearing surfaces of journal roller and thrust bearings are circular in cross-section and either cylindrical or conical in profile.

The accompanying table gives a classification of machinery bearings. Flat constraining surfaces for sliding machine parts having rectilinear movements have no common name, but are referred to as ways, guides, and the like.

The relative motion of all these constraining members and the members constrained is resisted at their surfaces of contact by a force which is called the force of friction. Thus, all kinds of bearings must be designed with particular reference to minimizing this force, and our starting-point must be a study of friction.

			Bearing Surfaces
Bearings with sliding contact		Journal	Cylindrical. Conical. Spherical.
		Thrust	Flat. Conical. Spherical. Generated by curve of tractrix.
Bearings with rolling contact	Ball bearings	Radial	Spherical (of balls) with double curved surfaces (of races). Spherical (of balls) with conical (of races).
		Thrust	Spherical (of balls) with flat (of seats). Spherical (of balls) with double curved (of seats).
	Roller bearings	Journal	Cylindrical with cylindrical. Conical with conical.
		Thrust	Cylindrical with flat. Conical with conical.

The cross-section, perpendicular to the axis of rotation, of all these bearing surfaces is necessarily circular.

PART I

BEARINGS WITH SLIDING CONTACT

SECTION I

FRICTION OF BEARINGS WITH SLIDING CONTACT

All sliding surfaces, no matter how carefully they have been prepared, are known to consist of minute humps and hollows; this is true even of the smoothest surfaces that can be made although, of course, the height of the depressions and hollows varies with different kinds of materials and different degrees of finish. This is similar to, though much less in degree than the "nap" of woolen cloth and the "pile" of velvet. It may also be likened in an exaggerated degree to the "fur" on a rough planed or milled surface of cast iron.

When two solid surfaces are held in contact by an appreciable force, these minute parts of the surfaces interlock and resist relative motion. This resisting force acts tangentially to the surface of separation, and is the force of friction.

In addition to this interlocking action of surfaces, there is another that takes place between surfaces that are very carefully fitted together and are in intimate contact. This action is called adhesion and still further resists relative motion. One of the best examples of this adhesion of accurately finished surfaces is the action of the Swedish gages that are now familiar to most engineers and machinists. The working surfaces of these gages have a splendid lapped finish. When two of these surfaces are rubbed into intimate contact they will adhere with a force having an intensity several times that of atmospheric pressure; that is, the force of adhesion between the surfaces tending to prevent separation by a direct pull is much greater than the force exerted by the pressure of the atmosphere upon the exposed surfaces. Experiments witnessed by the author showed that the force required to slide one of the gages upon another was about 2 1/2 lb. when the two were in intimate contact. The surfaces were 1 3/8 in. long by 3/8 in. wide, representing an area of about 1/2 sq. in. The upper gage was 1/2 in. thick, thus giving a weight of about 1 oz. This force of adhesion is but little understood, but this simple experiment shows how much it can be quantitatively between accurate surfaces.

Just as the intimate contact of the surfaces tends to increase the resistance of the relative motion, a separation of these surfaces by oil, grease or other lubricant reduces this resistance to a considerable extent. This fact will lead us later on to a consideration of the lubrication of bearings.

In some machine elements, as clutches and brakes, this resistance to relative

motion is a desirable thing; but in bearings, on the contrary, it is undesirable as it requires power to overcome it and, if excessive, may cause the surfaces to cut and ruin the bearings. Thus a knowledge of the laws of friction is absolutely essential to the design of successful bearings.

Unfortunately, these laws are not thoroughly understood, and it is now recognized that many of the older and accepted principles only hold true for a very limited range of conditions.

FRICTION OF REST AND MOTION

Experiment has shown that it requires a greater force to start one surface to sliding over another than to continue this sliding after motion has begun. Therefore, two forces of friction are recognized; the friction of rest (sometimes called stiction) and the friction of motion. A knowledge of the laws of the friction of motion is of more importance and for that reason the friction of rest will be treated briefly.

Again, the laws of friction naturally divide themselves into two groups referred to the condition of the surfaces in contact; (a) for dry or unlubricated surfaces, (b) for lubricated surfaces.

COEFFICIENT OF FRICTION

In a preceding paragraph it is stated that the force of friction is the resisting force to relative motion when two solid surfaces are held in contact by an appreciable force. The ratio of the force required to produce motion to the force holding the surfaces in contact is called the *coefficient of friction.*

For flat surfaces, if we let F equal the frictional resistance, W the normal load, and f the coefficient of friction, then

$$F=fW, \text{ or } f=\frac{F}{W}.$$

For flat surfaces where the top body is acted upon by its own weight only, the coefficient of friction is equal to the tangent of the angle of repose; that is, the angle of inclination with the horizontal of the plane of contact of the two surfaces where sliding will begin to take place. If this angle is indicated by a, the coefficient of friction equals the tangent of angle a. Or, $f=\tan. a$.

In a similar manner the coefficient of friction for cylindrical bearings is the ratio of the frictional resistance to the normal pressure and is again indicated by f. The character and fit of the surfaces, however, have a great deal to do with the value of the coefficient for such cases; the distribution of the normal pressure is variable, and the intensity of the normal pressure is a difficult quantity to

determine accurately. For convenience, it is customary to take as the normal pressure the intensity of pressure per unit of projected area of the bearing. Thus, if d is the diameter of the shaft and l its length, the intensity of pressure per unit of projected area is

$$w=\frac{W}{dl}.$$

The reason for the difficulty in determining accurately the intensity of normal pressure is evident to anyone familiar with bearings. It is perfectly possible to fit a journal into its bearing so that there shall be a complete arc of contact for 180 degrees of bearing circumference. In practice, however, a journal is made free in its bearing, the amount of freeing depending upon circumstances. Thus the arc of contact is decreased. In many cases this arc may not be over one-half of the total bearing depth of the journal, or say about 120 degrees. Again, journals and bearings tend to wear in use and thus produce a varying amount of contact depending upon the amount and location of the wear that has taken place.

WORK OF FRICTION

The work of friction is an important point to consider in the design of bearings, for it gives a measure of the amount of energy transformed into heat by the frictional resistance. This quantity of heat has a direct bearing upon the performance and life of many types of bearings. In some cases, a circulation of water or oil is established to conduct away the heat thus liberated.

For flat plates H the foot pounds of energy absorbed per minute equals $f W v$, where v is the velocity in feet per minute and W is expressed in pounds. For circular surfaces, if u is the number of revolutions per minute

$$H=\frac{f\pi dnW}{12}=0.2618\,fdnW.$$

Table 1 gives the moment of friction in inch pounds and the energy lost in friction in foot pounds per minute for the ordinary types of machinery bearing surfaces. The notation is as follows:

f = Coefficient of friction.
W = load on journals or pivots in pounds.
r = radius in inches.
r_1 = inner radius of collar in inches.
r_2 = outer radius of collar in inches.
d = diameter in inches.
n = number of revolutions per minute.
b = half of the included angle of cone.

Kind of bearing surfaces	Moment of friction in inch pounds	Energy lost in friction in foot pounds per minute
Journal bearings as ordinarily fitted	$0.5\,fdW$	$0.2618\,fdWn.$
Journal bearings tightly fitted	$0.635\,fdW$	$0.3324\,fdWn.$
Journal bearings fitted to grasp the shaft and give a uniform pressure throughout.	$0.78\,fdW$	$0.4112\,fdWn.$
Flat pivot bearings	$0.66\,frW$	$0.349\,fWrn.$
Flat collar bearings	$0.66\,fW\frac{r_2^3-r_1^3}{r_2^2-r_1^3}$	$0.349\,fWn\frac{r_2^3-r_1^3}{r_2^2-r_1^2}$
Conical pivot bearings	$0.66\,fWr\ cosec.\ b.$	$0.349\,fWrn\ cosec.\ b.$
Conical journal bearings or tapered journal bearings.	$0.66\,fWr\ sec.\ b$	$0.349\,fWrn\ sec.\ b.$
Truncated cone pivot bearings	$0.66\,fW\frac{r_2^3-r_1^3}{r_2\ sin.\ b}$	$0.349\,fWn\frac{r_2^3-r_1^3}{r_2\ sin.\ b}$
Hemispherical pivot bearings	frW	$0.2618\,fdWn.$
Schiele's or anti-friction bearings (tractrix curve).	frW	$0.2618\,fdWn.$

TABLE 1.—MOMENT OF FRICTION AND WORK OF FRICTION FOR BEARINGS WITH SLIDING SURFACES.

The mathematical work showing the derivation of these formulas of Table 1 can be found in Thurston's *Friction and Lost Work*, beginning on page 40.

Attention should be directed to the three items in Table 1 dealing with journal bearings. The first is for bearings as ordinarily fitted; the second with bearings tightly fitted and shows that the loss of energy in friction is 1.27 times that of properly fitted bearings. The third deals with bearings that have been so fitted, or have so worn that the intensity of pressure is uniform throughout. Such a condition produces a loss of power 1.57 times that of properly fitted bearings.

POWER LOST IN BEARINGS

To obtain the horsepower lost in bearings, or the amount of power required to drive a journal or pivot against its own friction divide the quantities obtained

from the formulas in the last column of Table 1 by 33,000. Similarly, to obtain the number of British thermal units of heat liberated in a given bearing, divide the quantities obtained from the formulas in the last column of Table 1 by 778.

FRICTION OF UNLUBRICATED SURFACES

The early experimental work to determine the values of the coefficient of friction for various surfaces in contact was done by Coulomb, Morin and Rennie. The work of General Morin, a most original and scientific engineer, was the most extensive. From his experiments he deduced three laws, which are

1. The frictional resistance is proportional to the pressure.
2. The frictional resistance is independent of the speed.
3. The frictional resistance is independent of the extent of the surfaces.[1]

These laws have frequently been quoted and treated as if they were rigidly true. Such, however, is not the case. They are now known to be only approximately true for a very limited range of conditions. In fact, General Morin himself considered them only as approximations as shown by this paragraph from a letter written by him to the secretary of the Institute of Mechanical Engineers, England, and printed in the proceedings of that Institution for the year of 1883, at page 663. "The results furnished by my experiments as to the relations between pressure, surface, and speed, on the one hand, and sliding friction on the other, have always been regarded by myself, not as mathematical laws, but as the close approximations to the truth within the limits of the data of the experiments themselves. The same holds, in my opinion, for many other laws of practical mechanics; such as those of rolling resistance, etc."

Table 2 is translated from the work of General Morin and gives the results of some of his experiments on plain surfaces. The values are for static friction, or friction at very low velocity, with light pressures. The surfaces were dry, or but very slightly lubricated.

In Table 2 it will be noticed that the values in the last column are considerably greater than those in the first. This shows us the effect of leaving comparatively soft bodies in contact for a considerable length of time. The irregular particles of the surfaces tend to compress and interlock to a greater degree than if the surfaces are merely brought in contact and motion at once takes place.

The experiments of Rennie in 1829 were made in general with dry and unlubricated surfaces. The principles that he deduced were that the friction of sliding surfaces differs with the character of the surfaces; that the friction of lubricated surfaces depends upon the lubricant, rather than upon the bodies themselves; that friction is least with hard materials and greatest with soft ones;

[1] See Annales des Mines, 3 serie, October, 1836, p. 27.

Surfaces in contact	Mutual arrangement of the fibers	Coefficient of friction: When the body is in motion	Coefficient of friction: When the surfaces have been some time in contact
Oak on oak, dry	Parallel	0.48	0.60 a 0.65
Oak on oak, dry	Perpendicular	0.32	0.54
Oak on oak, wet	Perpendicular	0.25	0.71
Elm on oak, dry	Parallel	0.43	0.69
Elm on oak, dry	Perpendicular	0.45	0.57
Ash on oak, dry	Parallel	0.40	0.50
Fir on oak, dry	Parallel	0.36	0.52
Beech on oak, dry	Parallel	0.36	0.53
Wild pear tree on oak, dry	Parallel	0.40	0.44
Wrought iron on oak, dry	Parallel	0.62	0.62
Yellow copper on oak, dry	Parallel	0.62	0.62
Black curried leather on oak, dry	Parallel	0.27	0.74
Cowhide sole leather laid flat on oak, dry.	Parallel	0.52	0.61
Sole leather on edge on oak, dry	Parallel	0.34	0.43
Sole leather on edge on oak, wet	Parallel	0.29	0.79
Mat of small hempen cords on oak, dry.	Parallel	0.32	0.50

TABLE 2.—COEFFICIENTS OF FRICTION FOR VARIOUS SURFACES IN CONTACT (*Morin*).

that the limit of abrasion of two rubbing surfaces is determined by the hardness of the softer, and that with wood and metals friction varies with the pressure and is independent of the extent of the surface, time of contact and velocity. Table 3 is from Rennie's experiments.

Pressure in pounds per square inch	Brass on cast iron	Wrought iron on wrought iron	Wrought iron on cast iron	Steel on cast iron
187	0.23	0.25	0.28	0.30
224	0.22	0.27	0.29	0.33
336	0.21	0.31	0.33	0.35
448	0.21	0.38	0.37	0.35
560	0.23	0.41	0.37	0.36
672	0.23	abraded	0.38	0.40
784	0.23	abraded	abraded	abraded

TABLE 3.—COEFFICIENTS OF FRICTION OF UNLUBRICATED SURFACES (*Rennie*).

Tables 4 to 7, inclusive, are also from General Morin's experiments and were made at low velocities, with surface either dry, oily, or greasy, as stated in the tables. In the original table of General Morin's from which Table 5 has been translated, there is a column giving the position of the fibres of the wooden test pieces with reference to the direction of motion, and in some cases with reference to the supposed direction of the fibres of the metal test pieces. However, the references are not entirely clear and have been omitted in the translations; apparently all of the fibres of the wooden test pieces ran parallel with the direction of motion and parallel with the supposed direction of the fibers of the metal pieces. Similarly a column has been dropped in translating Table 6, with the belief that the values are not rendered less useful thereby. Inconsistencies that may be noted in the values of the coefficients of friction may be explained by variations in the conditions of the tests.

Surfaces in contact	Condition of the surfaces	Mutual arrangement of the fibers	Coefficient of friction
Oak on oak	Coated with dry soap	Parallel	0.164
Oak on oak	Coated with tallow	Parallel	0.075
Oak on oak	Coated with lard	Parallel	0.067
Oak on oak	Oily	Parallel	0.108
Oak on oak	Dry	Perpendicular	0.336
Oak on oak	Coated with tallow	Perpendicular	0.083
Oak on oak	Coated with lard	Perpendicular	0.072
Oak on oak	Oily	Perpendicular	0.143
Beech on oak	Coated with tallow	Parallel	0.055
Beech on oak	Oily	Parallel	0.153
Elm on oak	Coated with dry soap	Parallel	0.137
Elm on oak	Coated with tallow	Parallel	0.070
Elm on oak	Coated with lard	Parallel	0.060
Elm on oak	Oily	Parallel	0.119
Elm on elm	Coated with dry soap	Parallel	0.139
Elm on elm	Oily	Parallel	0.140
Oak on elm	Dry	Parallel	0.246
Oak on elm	Coated with dry soap	Parallel	0.136
Oak on elm	Coated with tallow	Parallel	0.073
Oak on elm	Coated with lard	Parallel	0.066
Oak on elm	Oily	Parallel	0.136

TABLE 4.—COEFFICIENTS OF FRICTION FOR WOOD ON WOOD, DRY OR SCANTILY LUBRICATED SURFACES (*Morin*).

Surfaces in contact	Condition of the surfaces	Coefficient of friction
Wrought iron on oak	Moistened with water	0.256
Wrought iron on oak	Coated with dry soap	0.214
Wrought iron on oak	Coated with tallow	0.085
Cast iron on oak	Dry	0.490
Cast iron on oak	Coated with dry soap	0.189
Cast iron on oak	Moistened with water	0.218
Cast iron on oak	Coated with tallow	0.078
Cast iron on oak	Coated with lard	0.075
Cast iron on oak	Coated with olive oil	0.075
Cast iron on oak	Oily	0.107
Copper on oak	Coated with tallow	0.069
Copper on oak	Oily	0.100
Cast iron on elm	Dry	0.195
Cast iron on elm	Coated with tallow	0.077
Cast iron on elm	Coated with olive oil	0.061
Cast iron on elm	Coated with lard and plumbago	0.091
Cast iron on elm	Greasy after coating with tallow	0.125
Cast iron on elm	Greasy after coating with lard and plumbago.	0.137
Wrought iron on elm	Dry	0.252
Wrought iron on elm	Coated with tallow	0.078
Wrought iron on elm	Coated with lard	0.076
Wrought iron on elm	Coated with olive oil	0.055
Wrought iron on elm	Oily	0.138
Oak on cast iron	Coated with tallow	0.080
Oak on cast iron	Oily	0.168
Elm on cast iron	Coated with tallow	0.066
Elm on cast iron	Oily	0.135
Lignum-vitæ on cast iron	Coated with tallow	0.074
Lignum-vitæ on cast iron	Coated with olive oil	0.076
Lignum-vitæ on cast iron	Oily	0.121
Oak on wrought iron	Coated with tallow	0.098
Oak on wrought iron	Oily	0.149
Lignum-vitæ on bronze	Coated with tallow	0.082
Lignum-vitæ on bronze	Coated with olive oil	0.053
Lignum-vitæ on bronze	Oily	0.146

TABLE 5.—COEFFICIENTS OF FRICTION FOR WOOD AND METAL, DRY OR SCANTILY LUBRICATED SURFACES (*Morin*).

Surfaces in contact	Condition of the surfaces	Coefficient of friction
Cast iron on cast iron	Dry	0.152
Cast iron on cast iron	Moistened with water	0.314
Cast iron on cast iron	Coated with soap	0.197
Cast iron on cast iron	Coated with tallow	0.100
Cast iron on cast iron	Coated with lard	0.070
Cast iron on cast iron	Coated with olive oil	0.064
Cast iron on cast iron	Coated with lard and plumbago	0.055
Cast iron on cast iron	Oily	0.144
Wrought iron on cast iron	Dry	0.194
Wrought iron on cast iron	Coated with tallow	0.103
Wrought iron on cast iron	Coated with lard	0.076
Wrought iron on cast iron	Coated on olive oil	0.066
Steel on cast iron	Dry	0.202
Steel on cast iron	Coated with tallow	0.105
Steel on cast iron	Coated with lard	0.081
Steel on cast iron	Coated on olive oil	0.079
Steel on cast iron	Oily	0.109
Yellow copper on cast iron	Dry	0.189
Yellow copper on cast iron	Coated with tallow	0.072
Yellow copper on cast iron	Coated with lard	0.068
Yellow copper on cast iron	Coated with olive oil	0.066
Yellow copper on cast iron	Oily	0.115
Bronze on cast iron	Dry	0.217
Bronze on cast iron	Coated with tallow	0.086
Bronze on cast iron	Coated with olive oil	0.077
Bronze on cast iron	Oily	0.107
Cast iron on wrought iron	Coated with tallow	0.098
Cast iron on wrought iron	Coated with lard	0.053
Cast iron on wrought iron	Coated with olive oil	0.063
Wrought iron on wrought iron	Dry	0.138
Wrought iron on wrought iron	Coated with tallow	0.082
Wrought iron on wrought iron	Coated with lard	0.081
Wrought iron on wrought iron	Coated with olive oil	0.070
Wrought iron on wrought iron	Oily	0.177
Steel on wrought iron	Coated with tallow	0.093
Steel on wrought iron	Coated with lard	0.076
Bronze on wrought iron	Dry	0.161
Bronze on wrought iron	Coated with tallow	0.081
Bronze on wrought iron	Coated with lard and plumbago	0.089
Bronze on wrought iron	Coated with olive oil	0.072
Bronze on wrought iron	Oily	0.166
Cast iron on bronze	Dry	0.147
Cast iron on bronze	Coated with tallow	0.085

TABLE. 6.—COEFFICIENTS OF FRICTION FOR METAL ON METAL, DRY OR SCANTILY LUBRICATED SURFACES (*Morin*).

Surfaces in contact	Condition of the surfaces	Coefficient of friction
Cast iron on bronze	Coated with lard	0.070
Cast iron on bronze	Coated with olive oil	0.067
Cast iron on bronze	Oily	0.132
Wrought iron on bronze	Dry	0.172
Wrought iron on bronze	Coated with tallow	0.103
Wrought iron on bronze	Coated with lard	0.075
Wrought iron on bronze	Coated with olive oil	0.078
Wrought iron on bronze	Oily	0.160
Steel on bronze	Dry	0.152
Steel on bronze	Coated with tallow	0.056
Steel on bronze	Coated with olive oil	0.053
Steel on bronze	Coated with lard and plumbago	0.067
Bronze on bronze	Dry	0.201
Bronze on bronze	Coated with olive oil	0.058
Bronze on bronze	Oily	0.134

TABLE 6 (*Continued*).—COEFFICIENTS OF FRICTION FOR METAL ON METAL, DRY OR SCANTILY LUBRICATED SURFACES (*Morin*).

Surfaces in contact	Condition of the surfaces	Arrangement of the surfaces	Coefficient of friction
Tanned cowhide on cast iron.	Dry	The leather laid flat	0.559
Tanned cowhide on cast iron.	Moistened and saturated with water.	The leather laid flat	0.365
Tanned cowhide on cast iron.	Coated with tallow	The leather laid flat	0.159
Tanned cowhide on cast iron.	Coated with olive oil	The leather laid flat	0.133
Tanned cowhide on cast iron.	The leather oily, the cast iron moistened with water.	The leather laid flat	0.229
Tanned cowhide on cast iron.	Moistened with water	The leather placed edgewise.	0.338
Tanned cowhide on cast iron.	Moistened with olive oil	The leather placed edgewise.	0.135

TABLE 7.—COEFFICIENTS OF FRICTION FOR LEATHER ON CAST IRON, DRY OR SCANTILY LUBRICATED SURFACES (*Morin*).

As speeds increase the value of the coefficient of friction decreases, but the amount of this decrease is unknown experimentally, except for certain combinations of metals. In *Elements of Machine Design* by Kimball and Barr,

page 99, the coefficient of friction for cast iron or steel at a velocity of 440 ft. per minute is given as 0.32. At 2460 ft. per minute as 0.2, and at 5280 ft., or a mile a minute, as 0.006. It is also stated that these data for cast iron on steel will serve as a rough guide to what may be expected to occur for other combination of materials.

A consideration of the values of the coefficient of friction in Table 2 show that a great decrease takes place as soon as motion begins, even although the velocity is very low. It might be supposed that the change is very abrupt from the friction of rest of the friction of motion, but it is now generally believed that this change is gradual, and that the value of the coefficient at rest is not far different from the value at a very slow speed.

LAWS OF DRY OR UNLUBRICATED FRICTION

John Goodman sums up the laws governing dry friction in this manner, *Proc. Inst. C. E.*, Vol. LXXXV, page 391.

1. The friction between dry surfaces under moderate loads and low velocities varies directly as the normal pressure between them.
2. The normal pressure remaining unchanged, the friction is independent of the area in contact.
3. The friction is always greater on the reversal of direction of sliding.
4. The friction sensibly diminishes with a rise of the temperature.

These laws do not differ materially from those of General Morin, except that they are restricted to low velocities and moderate loads. As a matter of fact, these were the conditions under which General Morin experimented.

FRICTION OF LUBRICATED SURFACES

The laws for lubricated friction must be considered from an entirely different standpoint than those from dry friction, although it is very difficult, if not impossible, to define accurately where one condition begins and the other ends. In a journal bearing, properly lubricated, the friction resistance must obviously conform to the laws for lubricated surfaces. On the other hand, if the supply of lubricant is intermittent, as the quantity diminishes the resistance will tend to conform to the laws for dry or unlubricated surfaces. This latter condition is represented by the usual oil hole and squirt-can method of oiling. Other familiar forms of oiling are by means of sight feed and syphon lubricators, pad lubricators, oiling rings and chains, and oil baths. The efficiency of these devices increases in the order named.

Lubricant is introduced between bearing surfaces in order to separate them by a film of lubricant and thereby change the frictional resistance from that

of the solid surfaces in contact to the fluid-resistance of the film. Thus the nature and condition of the lubricant has more to do with this kind of friction than the bearing surfaces themselves assuming that they are well fitted.

FRICTION OF REST OF LUBRICATED SURFACES

It is frequently observed that after a machine has been idle for some time, it "starts hard." The bearings are sometimes said to be "stiff." The reason for this is that the pressure of the journals in the bearings tends to squeeze out the lubricant and allows the metallic surfaces to come into more or less close contact. Experience has shown that it is very difficult even with small areas and heavy loads to squeeze out all the lubricant, but its thickness will be very much reduced if the bearing is idle for any considerable length of time. Thus when a machine is started after a long shut down the frictional resistance in its bearings is more nearly that of dry or unlubricated surfaces than of well lubricated surfaces. This fact shows the necessity of carefully oiling a heavy machine before it is started. If this is not done the surface of journals and bearings may be in such close contact that abrasion will take place before the lubricant remaining in the bearing can be distributed and an oil film established.

The coefficients of rest or starting of motion as given by Professor Thurston in *Friction and Lost Work*, page 319, for sperm oil, lard oil and mineral oil are, respectively: 0.15; 0.1; 0.15. These are very much higher than similar coefficients for well-lubricated surfaces when in motion. These values were obtained from experiments made on a testing machine at pressures ranging from 75 to 500 lb. per square inch and with lubricant between metallic surfaces.

FRICTION OF POORLY LUBRICATED SURFACES

In the preceding paragraph is pointed out the fact that the friction of lubricated surfaces at rest is much greater than the friction of similar surfaces well lubricated when in motion. Between these two extremes is the condition of imperfect lubrication. This is the condition under which most machine bearings are run. Yet there are so many variables that it is impossible to establish any laws. The better the imperfect lubrication, the more nearly will the conditions conform to the laws for well lubricated surfaces. The poorer the lubrication, the more nearly will these conditions conform to the laws of dry surfaces.

The variables that govern the condition of poorly lubricated surfaces are the character of the surfaces themselves, their fit, the supply and nature of the lubricant, the pressure between them, the velocity of rubbing and the temperature as affecting the viscosity of the oil or grease. With so many variables it is easy to see the difficulties of obtaining results from even the most careful experimental work.

Efforts have been made to state the laws of imperfect friction, but with very little success. It is probably of more value to consider briefly some of the actions that take place in a bearing.

The pressure tends to squeeze out the lubricant. If this pressure is very high it may be impossible to get a sufficient supply of oil between the surfaces to produce a proper film. Then the bearing would be practically in metallic contact with its journal, and the coefficient of friction would be nearly that of dry surfaces in contact.

Again, if the velocity of rubbing is so very low that the lubricant cannot be drawn between the surfaces to form a film, there may be practically metallic contact.

This same condition may be produced by an insufficient quantity of oil and this is the most general cause of imperfect lubrication. If the pressure and velocity of rubbing are normal the supply of oil or grease may be such as to give any condition of lubrication between that of unlubricated surfaces and well lubricated surfaces.

The influence of temperature is important, for even with a suitable supply of lubricant at a given pressure, velocity of rubbing and ordinary temperature, the thinning of the oil as the temperature increases may be sufficient to break down the film and produce metallic contact.

Excellent lubrication can only be obtained by a plentiful supply of oil, as by an oil bath, flooded lubrication, and the best arrangements of oiling rings. The greater number of machine bearings are oiled either by a squirt can through an oil hole, by sight-feed oil cups, syphon oil cups, pads, or compression grease cups. All of these devices as ordinarily used can be considered as producing nothing but imperfect lubrication, and thereby bear out the general statement that the great mass of machinery bearings are in this second indefinite condition of poorly lubricated surfaces.

The only way in which this subject can be satisfactorily considered is by studying its limiting factors. In a former paragraph, page 15, is discussed the friction of dry surfaces and the following treats of the friction of well lubricated surfaces.

WELL LUBRICATED SURFACES

The best condition for a bearing to be in, and unfortunately the rarest, is, of course, well lubricated. This means a plentiful supply of lubricant, as from an oil bath. When this condition prevails the surfaces of the bearings are definitely separated by an oil film and the frictional resistance is the resistance to shearing of this film due to the rubbing action. The journal floats on the lubricant.

As the pressure on the journal under ordinary conditions has a definite

direction, the space between the journal and bearing is less at the point of nearest approach than at a point diametrically opposite. This means that the space between the journal and bearing is a wedge-shaped ring. The action of the journal as it rotates is to take oil from the region of least pressure and carry it into the region of greater pressure. This action is sufficient to suck oil against small heads. This continual drawing in of oil preserves the film and at the same time necessitates the doing of a certain amount of work in transferring the oil from one point to another in the bearing and in overcoming its own fluid resistance.

We are fortunate in the amount of experimental work that has been done to determine the laws of friction for well lubricated surfaces. This subject is by no means in the haze that surrounds poorly lubricated surfaces, although much work still awaits any investigator who chooses to enter the field.

THEORY OF BEARING LUBRICATION

Professor Osborne Reynolds has given a most interesting and valuable theoretical discussion of bearing lubrication in a paper, "Theory of Lubrication," *Phil. Trans.* (Royal Society, London), 1886. This should be read by everyone who wishes to grasp the fundamentals of the subject, yet the empirical rules given in following paragraphs are more directly serviceable for designing purposes.

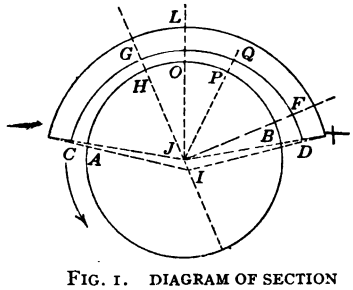


FIG. 1. DIAGRAM OF SECTION OF JOURNAL AND ITS MATING BEARING.

Professor Kingsbury, *Journal Am. Soc. Naval Engineers*, Vol. IX, No. 2, (1897) gives six laws for perfect lubrication arranged from Reynold's paper. The reference is to Fig. 1, reproduced from Reynolds and representing a right section of a cylindrical journal and its brass; the rotation of the journal is counter-clockwise. The conditions are such that the film is complete.

1. The vertical components of the pressure and friction balance the load.

2. The horizontal component of the oil pressure balances the horizontal component of the friction.

3. When the brass is unloaded, its point of nearest approach will be its middle point. The pressures are then symmetrically disposed with reference to O, the positive on the right, the negative on the left.

4. As the load increases, the positive vertical component must overbalance the negative component. This requires that H should be at the left of O.

5. As the load increases, OH reaches a maximum value which places H nearly, but not quite as the left extremity of the brass, but still leaves JI small

as compared with GH. For a further increase of the load, H moves back again toward O.

6. As the load increases, the distance between the axes of brass and journal increases.

LAWS OF FRICTION FOR WELL LUBRICATED SURFACES

The following five empirical laws of friction for well lubricated surfaces have been modified from four laws laid down by John Goodman, *Proc. Inst. C. E.*, Vol. LXXXV, page 380, and from the results of Lasche's experiments. It is, of course, uncertain just how far they apply. But each law is amplified by the results of experiments.

1. The coefficient of friction for well lubricated surfaces is practically independent of the materials composing the bearings and journals, but is dependent upon the nature of the lubricant.

2. The coefficient of friction for well lubricated surfaces is from 1/6 to 1/10 that for unlubricated or scantily lubricated surfaces.

3. The coefficient of friction for moderate pressures and speeds varies approximately inversely as the normal pressure; that is, the frictional resistance per unit of area is a constant assuming the speed constant. The total frictional resistance varies directly as the area.

Or, $$\frac{f_1}{f_2}=\frac{p_2}{p_1} \text{ and } f_1p_1=f_2p_2$$

4. The coefficient of friction for well lubricated surfaces for a given pressure is very high for low rubbing velocities, though decreasing rapidly as the velocity increases. For velocities from 100 to 500 ft. per minute it decreases, varying approximately as the *square roots* of the velocities; for velocities from 500 to 1600 ft. per minute it decreases at a lower rate, varying approximately as the *fifth roots* of the velocities; for velocities above 1600 ft. per minute, it is practically independent of the velocity.

Or, for velocities from 100 to 500 ft. per minute $\frac{f_1}{f_2}=\frac{\sqrt{V_1}}{\sqrt{V_2}}$

Or, for velocities from 500 to 1600 ft. per minute $\frac{f_2}{f_3}=\frac{\sqrt[5]{V_2}}{\sqrt[5]{V_3}}$

Or, for velocities over 1600 ft. per minute $f_3=f_4$

5. The coefficient of friction varies approximately inversely as the temperature up to a point just before abrasion takes place.

DISCUSSION OF THE FIRST LAW

That the metals forming the journal and bearing in well lubricated bearings have little influence on the coefficient of friction has been observed by many

investigators. In *Traction and Transmission* for January, 1903, Lasche describing some of his own experiments says:

"The results of numerous experiments go to prove that the metal forming the journal—nickel steel, ordinary Siemens-Martin steel, or mild steel—has no marked influence on the degree of friction or friction work."

And again: "The journal was placed in a gun-metal bushing; in one with white-metal lining and in one with a mercury amalgam lining. As in the case with the metal forming the journal the various kinds of bushings gave approximately the same results."

The reason for this is that the load is fluid-borne and so the frictional resistance is that of the oil film not of the metallic surfaces.

DISCUSSION OF THE SECOND LAW

The truth of the second law is shown by the following table taken from the work of Beauchamp Tower, *Proc. Inst. M. E.* for 1883, page 651.

Method of lubrication	Coefficient of friction	Comparative friction
Pad under journal	0.009	6.48
Oil bath	0.00139	1
Syphon lubricator	0.0098	7.06

In another experiment made by Mr. Tower with a load of 293 lb. per square inch and a journal speed of 314 ft. per minute with an oil bath, a coefficient of friction of 0.0016 was obtained. Under the same conditions with a pad the coefficient was 0.0097 which is six times the amount of the former. Under the same condition Mr. Stroudley obtained a coefficient of 0.00961 which is in very close agreement.

DISCUSSION OF THE THIRD LAW

The following tabulation is from the experiments of Mr. Tower, *Proc. Inst. M. E.* for 1883, page 650.

Load in lb. per sq. in.,	520	468	415	363	310	258	205	153	100
Frictional resistance per sq. in.,	0.416	0.514	0.498	0.472	0.464	0.438	0.43	0.458	0.45

This shows that the unit frictional resistance is sensibly constant with varying loads.

The frictional resistance per square inch is the product of the coefficient of friction times the load per square inch on the projected area of the bearing. Thus, if this product is a constant one factor must vary inversely as the other;

a high load will give a low coefficient and a low load a high coefficient. This is true for well lubricated surfaces.

For ordinary lubrication the coefficient is more nearly constant under varying loads; the friction resistance then varies directly as the load. This is shown by Table 8 taken from the results of the experimental work of Beauchamp Tower, *Proc. Inst. M. E.*, 1883.

Nominal load lb. per sq. in.	Actual load lb. per sq. in.	Temperature Fahr.	100 rev. 105 ft. per min. lb.	150 rev. 157 ft. per min. lb.	200 rev. 209 ft. per min. lb.	250 rev. 262 ft. per min. lb.	300 rev. 314 ft. per min. lb.	350 rev. 366 ft. per min. lb.	400 rev. 419 ft. per min. lb.
328	582	90°	.....	.....	3.5	3.35	3.21	.....	.....
310	551	82°	.....	3.06	3.06	2.84	3.06	.....	.....
293	520	76°	.....	3.06	3.06	2.84	2.84	.....	.....
275	498	77°	.....	2.49	2.49	2.62	2.84	.....	.....
258	458	78°	.....	2.89	2.44	2.28	2.17	2.1	2.13
205	364	82°	.....	2.145	1.78	1.735	1.605	1.75	2.04
153	272	74°	1.56	1.37	1.473	1.56	1.605	1.81	1.89
100	178	75°	1.0	0.992	1.093	1.225	1.33	1.44	1.54

The chord of the arc of contact of the brass = 2 1/4 in.

The nominal load per square inch is the total load divided by 4×6.

The actual load per square inch is the total load divided by 2 1/4×6.

The results with the actual load of 582 lb. per square inch were obtained with difficulty, and the bearing seized with that load after running for a short time.

The lubricating pad consisted of a piece of felt pressing against the journal, and resting on worsted immersed in a tin box full of oil.

TABLE 8.—RELATIONSHIP OF LOAD AND FRICTIONAL RESISTANCE FOR ORDINARY LUBRICATION

Mr. Stroudley with pad lubrication on railroad car axles on the Brighton Railway in England obtained the result of Table 9 which still further supports the law.

Load per square inch on brass in pounds	Coefficient of friction	Friction resistance per square inch in pounds
333	0.00792	2.639
293	0.0080	2.34
251	0.0077	1.93
237	0.0072	1.70

TABLE 9.—RELATIONSHIP OF LOAD AND FRICTIONAL RESISTANCE AS DETERMINED BY STROUDLEY.

DISCUSSION OF THE FOURTH LAW

Turning to the fourth law, A. M. Wellington, *Trans. A. S. C. E.*, Vol. XIII, found from experiment on journals revolving at very low velocities that the friction under those conditions was very great and nearly constant under widely varying conditions of lubrication, load and temperature. As the speed increased the friction fell slowly and regularly, returning again to the original amount when the velocity was reduced to the same rate. The tabulation below shows some of the results.

Feet per minute,	2.16	3.33	4.86	8.82	21.42	35.37	53.01	89.28	106.02
Coefficient of friction,	0.118	0.094	0.0705	0.0685	0.055	0.047	0.04	0.035	0.3 0.0255

This shows a regular decrease in the value of the coefficient of friction with increase of speed.

The results of experiments made by Professor Kimball, see page 1904 *American Journal of Science*, March, 1878, show this same law of the decrease of the coefficient of friction with the increase of speed. The following gives some of Professor Kimball's results:

Velocity in ft. per min.,	1	3	5	7	10	15	20	30	40	60	80	100
Coefficient,	0.15	0.122	0.114	0.093	0.079	0.066	0.058	0.054	0.053	0.052	0.051	0.05

These experiments were made with journal bearings and show that the friction was reduced 67 per cent. with an increase of velocity from 1 to 100 ft. per minute. After this limit of velocity 100 ft. per minute has been passed, the coefficient of friction varies approximately with the square root of the velocity.

Origin of data	Load lb. per sq. in.	Velocity in feet per minute					
		209	262	314	366	419	471
Observed.....	520	0.001	0.0012	0.0013	0.0014	0.0015	0.0017
Calculated.....			0.0118	0.00123	0.00132	0.00141	0.0015
Observed.....	468	0.0013	0.0014	0.0015	0.0017	0.0018	0.002
Calculated.....			0.00145	0.00159	0.00172	0.00184	0.00195
Observed.....	415	0.0014	0.0015	0.0017	0.0019	0.0021	0.0024
Calculated.....			0.00157	0.00172	0.00188	0.00198	0.0021

TABLE 10.—COMPARISON OF OBSERVED AND CALCULATED COEFFICIENTS OF FRICTION FOR VELOCITIES OVER 100 FT. PER MINUTE.

Table 10 shows us this fact. The observed coefficients of friction for various velocities are taken from some of Mr. Tower's experiments. The comparable calculated coefficients were computed from the ratio of the square roots of the velocities. The agreement is very close.

For the experimental work to determine the coefficient of friction at high speeds, we are indebted to Thurston, Stribeck and Lasche. In *Traction and Transmission*, January, 1903, Lasche thus sums up the results of these experiments: From 500 ft. per minute to 800 ft. per minute the experiments of Stribeck show that the rise in the value of the coefficient of friction is slow, and approaches the result of the Thurston experiment expressed in the formula

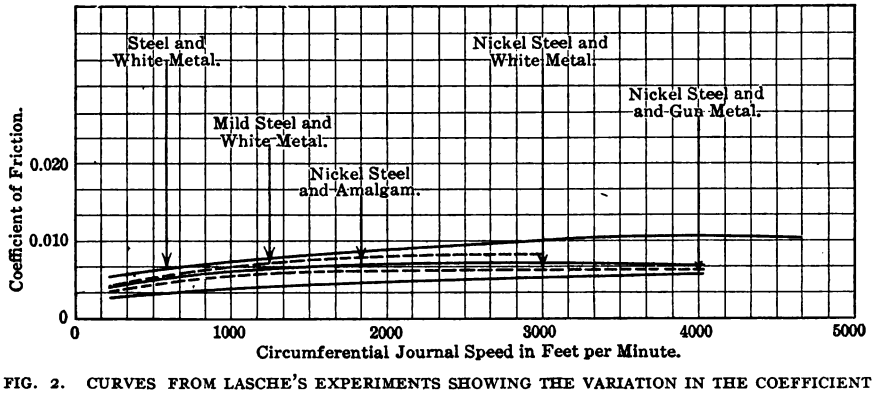


FIG. 2. CURVES FROM LASCHE'S EXPERIMENTS SHOWING THE VARIATION IN THE COEFFICIENT OF JOURNAL FRICTION WITH VELOCITY.

$f=\sqrt{V}$. At a still higher speed the influence of the velocity disappears and at 1600 ft. per minute the curve of the values of the coefficient is only slightly higher than at 800 ft. per minute. The experiments of Lasche further give the conclusion that for velocities of 2000 ft. per minute and over the coefficient of friction is practically independent of the velocity.

There is a break in these results between a speed of 1600 ft. per minute and 2000 ft. per minute, but from a consideration of the plotted curve of Lasche's work we seem justified in assuming that for velocities of 1600 ft. per minute and over the coefficient of friction is practically independent of the velocity.

Fig. 2 shows the general shape of the coefficient of friction curves plotted against velocities of rubbing as abscisses.

DISCUSSION OF THE FIFTH LAW

Law 5 states that the variation of friction with temperature is approximately in inverse ratio. The following taken from Mr. Tower's experiments at a velocity of 262 ft. per minute with an oil bath show this law:

Temperature in Fahrenheit degrees	Observed coefficient of friction	Calculated coefficient of friction
110	0.0044	0.00451
100	0.0051	0.00518
90	0.006	0.00608
80	0.0073	0.00733
70	0.0092	0.00964
60	0.0119	0.01252

However, this law does not hold good for imperfect lubrication, that is, for pad or syphon lubrication. Then the coefficient of friction diminishes more rapidly for a given temperature according to a gradually decreasing scale, until a normal temperature is reached. This is shown by the result of Mr. Stroudley's experiments with a pad of rape-seed oil.

Temperature Fahrenheit degrees	Coefficient of friction	Decrease of coefficient
105	0.022	0.004
110	0.018	0.002
115	0.016	0.002
120	0.014	0.0015
125	0.0125	0.001
130	0.0115	0.0005
135	0.011	0.0004
140	0.0106	0.0002
145	0.0102	.....

THICKNESS OF THE OIL FILM

A series of experiments made by Goodman to determine the thickness of the oil film showed that with pad lubrication on first starting the journal, the thickness of the film was 0.0013 in. As the speed increased and the film became better developed its thickness increased to 0.0029 in.

In the *American Machinist* for September 17, 1903, page 1316, Herbert S. Moore, writing in regard to a film formed with a heavy engine oil, states:

"It was at first attempted to get a perfect film in the Olsen-Carpenter friction machine, in which the test bearing was a snug fit all around the journal. It was found impossible to get such a film except at rare intervals of short duration, whereas with the altered Thurston machine, in which the diameter of the

bearing was 0.003 in. greater than that of the journal, this film was always readily formed and preserved."

If the film was uniform its thickness would thus be 0.0015 in.

BREAKING-DOWN POINT OF THE OIL FILM

The difficulty of lubricating bearings under high pressures is well known. The reason is evident. As the pressures are increased the film of oil is established and maintained with increasing difficulty. If the pressures are increased to a sufficient amount the film of oil finally breaks down, and the journal and bearings come into metallic contact.

Herbert S. Moore, on page 1283, *American Machinist* for September 10, 1903, gives a formula based on experimental investigations, to determine the pressure at which a film of machinery oil will break down. This formula is

$$p = 7 \cdot 47\sqrt{v}$$

in which p is the intensity of the load in pounds per square inch, and v is the rubbing velocity of the journal in feet per minute.

The experimental work consisted in connecting into an electric circuit the rotating journal and its bearing. As long as the oil film was properly maintained the resistance of this circuit was very high. As soon, however, as the oil film broke down, giving metallic contact between journal and bearing, the resistance of the circuit fell to almost nothing.

PRESSURE OF THE OIL FILM

If a journal is completely fluid-borne it follows that the oil film must be under a considerable pressure in order to support the weight of the journal and its attached parts. That such is a fact was demonstrated by an experiment by Mr. Towers reported in the *Proc. Inst. M. E.* for 1885, page 58. In these experiments a half bearing or brass was used, having a length of 6 in. and a chord length of the arc of contact of 3.9 in. Three longitudinal holes were drilled a little more than half way of the length of the bearing, and spaced as indicated by the letters *a*, *d* and *g* in Fig. 3.

At 9 points, as indicated by the letters in the figure, holes were drilled from the inside of the bearing into these three longitudinal channels. Each channel was connected with a pressure gage, and one hole at a time was left open from the channel into the inside of the bearing. By this means pressure readings were taken at nine different points, three on the side of the bearing where the journal entered, three on the middle line, and three on the side of the bearing where the journal left. Furthermore, as six of these holes were between a middle transverse plane and the end, and since it is fair to assume that the

same pressure would exist on the other side of this middle plane, it can be said that the experiments covered 15 points on the surface of the bearing.

The lubrication was by means of an oil bath, the journal being about half submerged. The speed of rotation was kept uniform at 150 revolutions per minute, and the temperature was held constant at 90° F. The total load on the bearing was 8008 lb., and the observed oil pressure at the 9 points investigated were as shown by the following:

For longitudinal plane at entering side of the journal *a* 310, *b* 335, *c* 370; for middle longitudinal plane *d* 565, *e* 615, *f* 625; for longitudinal plane at leaving side of the journal *g* 430, *h* 480, *i* 500.

A summation of the total pressure on the bearing from these observations gives 7988 lb. This compares very closely with the actual pressure, which was 8008 lb., the small difference being probably due to inaccuracies in observations.

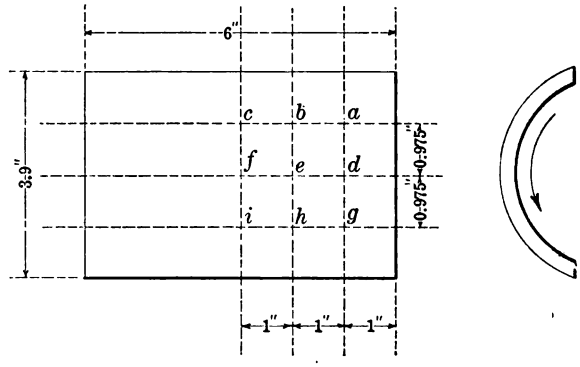


FIG. 3 DIAGRAM SHOWING POINTS AT WHICH OIL PRESSURES WERE TAKEN.

These results present several interesting facts. The length of the bearing was 6 in., and the length of its chord of contact 3.9 in., giving a projected area of 23.4 sq. in. The applied load was 8008 lb., which divided by the projected area gives an intensity of pressure per square inch of about 311 lb. The highest observed pressure at the point *f*, or center of the bearing area, was 625 lb., or double the computed intensity of pressure.

The observed pressures at the points *g*, *h* and *i* on the leaving side of the bearing are considerably greater than the corresponding pressures at the points *a*, *b* and *c* on the entering side. This shows the shifting of the point of maximum pressure in the bearing from a point diametrically below or above the axis of rotation, to a point with the direction of rotation. See the following section devoted to a discussion of the points of maximum and minimum pressure in an oil film.

The existence of these pressures in an oil film emphasizes the necessity of using good care and judgment in selecting the points where lubricant is to be introduced. Obviously oil cannot be introduced into a bearing unless it is under a greater pressure than the pressure in the oil film at the point of entry. Some failures in elaborate oiling systems can be traced to this very fact; the attempt to feed oil at a point in the bearing where the oil film was under a greater pressure than the head of the feed.

Referring again to Mr. Tower's experiment, he found that by varying the load the oil pressures rose and fell in exact proportion. Putting on weight increased the oil pressures, and removing weight decreased them. To determine the effect of speed on this oil pressure the journal speed was reduced from 150 revolutions per minute to 20, or as the diameter of the shaft was 4 in., from about 150 ft. per minute to about 20 ft. per minute. The gage pressures of the oil film were exactly the same at the lower speeds as at the higher. Reynolds' theory and Kingsbury's experiments show, however, that the distribution of the pressure does vary with speed.

As there are points of maximum pressure in the oil film in a bearing, there are likewise points of minimum pressure, and these may be below the pressure of the atmosphere, or at a slight vacuum. This applies only to completely circular bearings. At page 1282 of the *American Machinist*, September 10, 1903, Herbert F. Moore describes an experiment with a small dynamo bearing, in which the bearing sucked oil against a head of 6 in. We quote: "By simply leading a pipe from a reservoir of oil to that part of the bearing where the pressure would be negative, if anywhere, it is found that the bearing would suck in oil from a reservoir 6 in. below and thus lubricate itself."

POINTS OF MAXIMUM AND MINIMUM PRESSURE AND NEAREST APPROACH

In the earlier discussions of the relative positioning of a journal and its mating bearing when the journal was in motion with ample lubrication present, it was stated that the journal rolled back so that the point of nearest approach was on the *on* side of the bearing and at an angular distance from the vertical—assuming the load to act vertically with the bearing supporting the journal—equal to the angle of repose of the system. This theory has been shown to be fallacious by the experimental work of Tower and analytical work of Reynolds.

In an ordinary journal bearing where there is some looseness, when the journal is at rest it lies at the bottom of the bearing and the point of nearest approach, in this case contact, lies in a vertical passing through the centers of both. The rolling action outlined in the paragraph above may take place during the

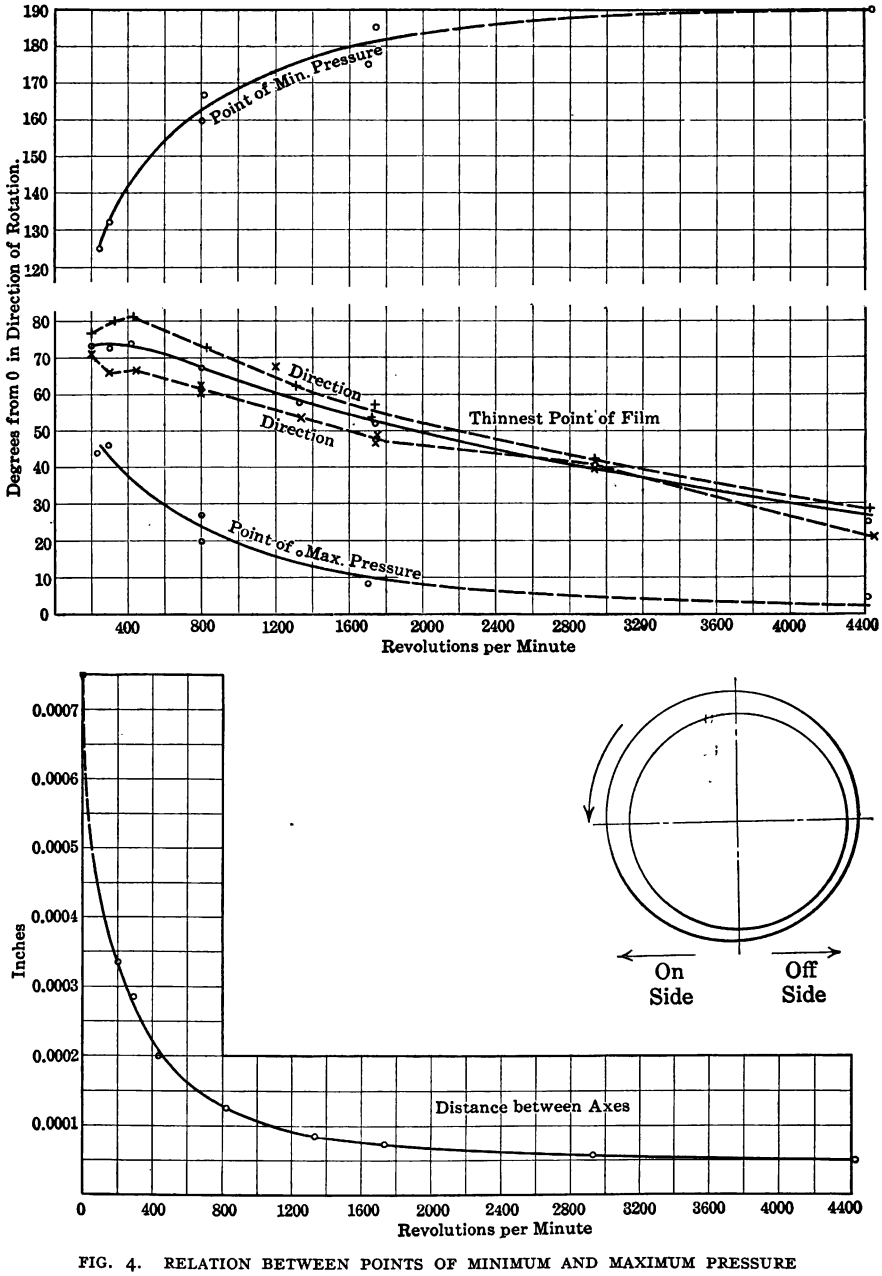


FIG. 4. RELATION BETWEEN POINTS OF MINIMUM AND MAXIMUM PRESSURE AND POINT OF NEAREST APPROACH.

period of starting, but if there is any appreciable amount of lubricant present, as soon as sliding takes place the point of nearest approach and the point of maximum pressure (which are *not* identical) are thrown to the *off* side of the vertical through the centers of journal and bearing. See the upper diagram in Fig. 4.

The angular amount of this shifting is dependent on the fit and finish of the surfaces, the quality of lubrication, the speed, and the pressure. With a variable load and constant speed, as the load increases the point of nearest approach swings upward continuing until it reaches a position nearly, but not quite, at the extremity of the horizontal axis. With a further increase of load, it swings back to a position found by Reynolds to be about 40 degrees from the vertical. With a still further increase of load, the oil film is ruptured and the conditions are changed from that of perfect lubrication to that of imperfect lubrication.

Quoting from Reynolds' paper, *Phil. Trans.* (Royal Society, London) 1886:

"The circumstances which determine the greatest load which a bearing will carry with complete lubrication, that is, with a film of oil extending between brass and journal throughout the entire area, are definitely shown in the theory.

"The effect of increasing the load beyond a certain small value being to cause the brass to approach nearer to the journal at a point on the off side which moves toward the vertical as the load increases and when the load is such that the least separating distance is about half the difference of the radii, the angular position of the point of nearest approach is 40 degrees to the off side of the vertical through the middle of the brass. At this point the pressure in the oil film is everywhere greater than at the extremities of the brass, but when the load further increases the pressure toward the extremity on the off side becomes smaller or negative. This, when sufficient, will cause rupture in the oil film, which will then only extend between the brass and journal over a portion of the whole area and a smaller portion as the load increases. Thus since the amount of negative pressure which the oil will bear depends on circumstances which are uncertain, the limit of the safe load for complete lubrication is that which causes the least separating distance to be half the difference of the radii of brass and journal."

An examination of Reynolds' equations shows that the load and speed are inversely related; that is, an increase of load has an effect upon the form of the film of the same kind as a decrease of speed, and *vice versa*.

That practice agrees with this theory is shown by a phenomenon observed by Tower and by the position of the wear in car journal brasses. Tower mentions that the journal having been run in one direction until the initial tendency to heat had entirely disappeared, on being reversed immediately began to heat again; but this effect stopped when the process had been often repeated. The fact being that running in one direction, the brass had been worn to the journal

only on the off side for that direction, so that when the motion was reversed the new off side acted like a new brass.

In car brasses the point of wear is always forward or on the off side.

The relative relations of the points of maximum pressure, minimum pressure, and nearest approach, are shown by the curves of Fig. 4. These are taken from a paper of Professor Kingsbury, published in the *Journal of the American Society of Naval Engineers*, Vol. IX, No. 2, 1897, and apply to an experiment with an air-lubricated bearing, running under a constant load and at variable speed.

SECTION II

COEFFICIENTS OF FRICTION OF JOURNAL, COLLAR, STEP AND GUIDE BEARINGS

From the discussion of imperfect lubrication, that is the ordinary lubrication to machinery journal bearings, it is evident that no fixed values can be given for the coefficient of friction. Any values that are given must be for definite lubricants, and not for surfaces in contact. With perfect lubrication and moderate velocity the coefficient may be as low as 0.001. With very low velocities and high pressures the coefficient approaches that for greasy or unlubricated surfaces, a minimum value for which may be taken as 0.15. Tests of new well-fitted bearings have given values of 0.30 and more. These values are the ones to use in estimating the starting moment for heavy machinery.

On page 105 of *Elements of Machine Design*, by Kimball and Barr, the coefficients of friction to be used in the design of ordinary machinery are given as follows: A fair average range for pressures from 50 to 500 lb. and velocities from 50 to 500 ft. per minute is from 0.02 to 0.008, and for purposes of design of ordinary machinery, may be taken at 0.015.

Perhaps the lowest coefficient of friction that has been reported from authentic tests will be found on page 149, *Trans. A. S. M. E.*, Vol. LXXIV. There Professor Kingsbury presents a table giving coefficients of friction for sperm oil under a pressure of 340 lb. per square inch at a temperature of 90° F., and varying rubbing speeds.

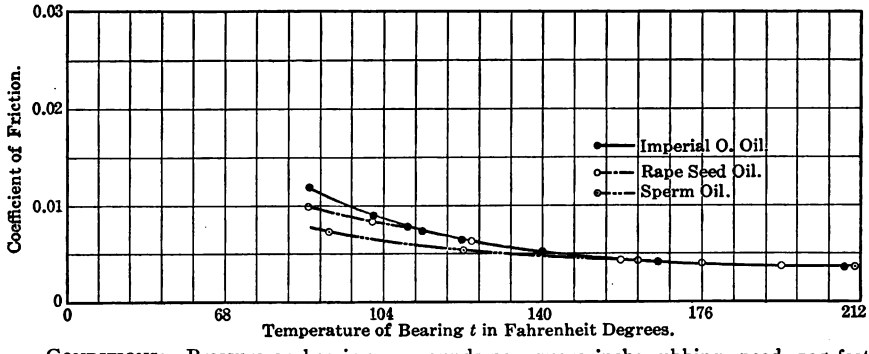
42 feet per minute	64 feet per minute	101 feet per minute
0.000906.....	0.001171	0.00146
0.000915.....	0.001158	0.00155
0.000886.....	0.001158	0.00151
0.000915.....	0.001125	.....

On page 146 of the same paper a value of 0.00053 is given.

The lowest coefficient of which the author has learned is 0.00046. This was also obtained by Professor Kingsbury in a test, the results of which have never

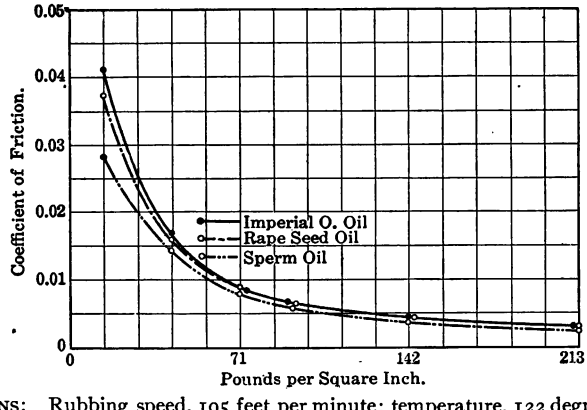
before been published. The lubricant was spindle oil, pressure 340 lb. per square inch, temperature 135° F., and a rubbing speed of 42.4 ft. per minute.

To give a definite idea of the value, the coefficients of friction for general bearings and the way in which they vary under different conditions, Tables 11 to 19, inclusive, are presented. They are all from the experiments of Mr. Tower, made under the auspices of the Institution of Mechanical Engineers. In using



CONDITIONS: Pressure on bearing, 92 pounds per square inch; rubbing speed, 195 feet per minute; Bearing flooded with oil, about 0.8 quart per minute; journal of nickel steel, bushes of white metal, diameter about 10 inches, length about 4 1/2 inches.

FIG. 5. COEFFICIENTS OF FRICTION FOR DIFFERENT LUBRICANTS (*Lasche*).



CONDITIONS: Rubbing speed, 195 feet per minute; temperature, 122 degrees Fahrenheit; bearing flooded with oil, about 0.8 quart per minute; journal nickel steel, bushings white metal, diameter about 10 inches, length about 4 1/2 inches.

FIG. 6. COEFFICIENTS OF FRICTION FOR DIFFERENT LUBRICANTS (*Lasche*).

any values from these tables it must be clearly recognized for what kind and degree of lubrication they are applicable.

Figs. 5 and 6 are reproduced from *Traction and Transmission*, January, 1903, and show results of Lasche's experiments. The "imperial oil" there mentioned is a Russian mineral oil of good quality for lubricating purposes.

Nominal load lb. per sq. in.	Speeds						
	100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min.
Lb.							
625	.....	0.0013	0.00139	0.00147	0.00157	0.00165	.....
520	.....	0.00123	0.00139	0.0015	0.00161	0.0017	0.00178
415	.....	0.00123	0.00143	0.0016	0.00176	0.0019	0.002
310	.....	0.00142	0.0016	0.00184	0.00207	0.00225	0.00241
205	0.00178	0.00205	0.00235	0.00269	0.00298	0.00328	0.0035
100	0.00334	0.00415	0.00494	0.00557	0.0062	0.00676	0.0073

CONDITIONS: A 4-in. journal of steel 6 in. long fitted on its upper side with a gun metal brass, or half box, embracing somewhat less than one-half the journal circumference—the chord of the arc contact equal to 3.92 in.—and when running kept at a constant temperature of 90° F.

TABLE 11.—COEFFICIENTS OF FRICTION FOR A BATH OF MINERAL OIL—TEMPERATURE CONSTANT (*Tower*).

Nominal load lb. per sq. in.	Speeds							
	100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min.	450 rev. 471 ft. per min.
Lb.								
625	.....	0.001	0.0012	0.0014	0.0014	0.0016	0.0018	0.002
520	.....	0.0014	0.0016	0.0018	0.0019	0.002	0.0021	0.0022
415	.....	0.0016	0.0019	0.0021	0.0023	0.0025	0.0026	0.0027
310	0.002	0.0022	0.0026	0.0029	0.0032	0.0035	0.0038	0.004
205	0.0026	0.0034	0.0040	0.0047	0.0053	0.0058	0.0062	0.0066
153	0.0028	0.0038	0.0048	0.0057	0.0065	0.0071	0.0077	0.0083
100	0.0054	0.0076	0.0094	0.0109	0.0123	0.0133	0.0142	0.0151

CONDITIONS: A 4-in. journal of steel 6 in. long fitted on its upper side with a gun metal brass, or half box, embracing somewhat less than one-half the journal circumference—the chord of the arc contact equal to 3.92 in.—and when running kept at a constant temperature of 90° F.

TABLE 12.—COEFFICIENT OF FRICTION FOR A BATH OF MINERAL GREASE—TEMPERATURE CONSTANT (*Tower*).

Nominal load lb. per sq. in.	Speeds							
	100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min.	450 rev. 471 ft. per min.
Lb.								
520	Seized	.....	.....	.....	.....	.....	.....	.....
415	.....	0.0015	0.0017	0.0018	0.0019	0.002	0.0021	0.0021
310	.....	0.0011	0.0012	0.0014	0.0016	0.0017	0.0018	0.0019
205	0.0013	0.0016	0.0018	0.0021	0.0023	0.0024	0.0025	0.0027
153	0.0016	0.0019	0.0023	0.0028	0.0030	0.0033	0.0035	0.0037
100	0.0025	0.003	0.0038	0.0044	0.0051	0.0057	0.0061	0.0064

CONDITIONS: A 4-in. journal of steel 6 in. long fitted on its upper side with a gun metal brass, or half box, embracing somewhat less than one-half the journal circumference—the chord of the arc contact equal to 3.92 in.—and when running kept at a constant temperature 90° F.

TABLE 13.—COEFFICIENT OF FRICTION FOR A BATH OF SPERM OIL—TEMPERATURE CONSTANT (*Tower*).

Nominal load lb. per sq. in.	Speeds							
	100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min.	450 rev. 471 ft. per min.
Lb.								
520	.....	0.0009	0.001	0.0011	0.0013	0.0015	0.0015	0.0017
415	.....	0.0012	0.0014	0.0015	0.0016	0.0018	0.0019	0.0021
310	.....	0 0014	0.0017	0.002	0.0022	0.0025	0.0026	0.0029
205	0.0017	0.0020	0.0023	0.0028	0.0031	0.0034	0.0039	0.0042
153	0.0022	0.0027	0.0032	0.0037	0.0041	0.005	0.0051	0.0052
100	0.0035	0.0042	0.005	0.006	0.0067	0.0076	0.0081	0.009

CONDITIONS: A 4-in. journal of steel 6 in. long fitted on its upper side with a gun metal brass, or half box, embracing somewhat less than one-half the journal circumference—the chord of the arc contact equal to 3.92 in.—and when running kept at a constant temperature of 90° F.

TABLE 14.—COEFFICIENTS OF FRICTION FOR LARD OIL BATH—TEMPERATURE CONSTANT (*Tower*).

Temperature Fahr.	Speeds							
	100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min.	450 rev. 471 ft. per min.
120°	0.0024	0.0029	0.0035	0.004	0.0044	0.0047	0.0051	0.0054
110°	0.0026	0.0032	0.0039	0.0044	0.005	0.0055	0.0059	0.0064
100°	0.0029	0.0037	0.0045	0.0051	0.0058	0.0065	0.0071	0.0077
90°	0.0034	0.0043	0.0052	0.006	0.0069	0.0077	0.0085	0.0093
80°	0.004	0.0052	0.0063	0.0073	0.0083	0.0093	0.0102	0.0112
70°	0.0048	0.0065	0.008	0.0092	0.0103	0.0115	0.0124	0.0133
60°	0.0059	0.0084	0.0103	0.0119	0.013	0.014	0.0148	0.0156

CONDITIONS: A 4-in. journal of steel 6 in. long fitted on its upper side with a gun metal brass, or half box, embracing somewhat less than one-half the journal circumference—the chord of the arc contact equal to 3.92 in.—and when running kept at a constant temperature of 90° F.

TABLE 15.—COEFFICIENTS OF FRICTION FOR BATH OF LARD OIL—LOAD CONSTANT AT 100 LB. PER SQUARE INCH (*Tower*).

Nominal load lb. per sq. in.	Speeds							
	100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min.	450 rev. 471 ft. per min.
Lb.								
520	.....	0.0008	0.001	0.0012	0.0013	0.0014	0.0015	0.0017
468	.....	0.0011	0.0013	0.0014	0.0015	0.0017	0.0018	0.002
415	.....	0.0012	0.0014	0.0015	0.0017	0.0019	0.0021	0.0024
363	.....	0.0013	0.0016	0.0017	0.0019	0.002	0.0022	0.0025
310	.....	0.0015	0.0017	0.0019	0.0021	0.0022	0.0024	0.0027
258	0.0014	0.0017	0.002	0.0023	0.0025	0.0026	0.0029	0.0031
205	0.0018	0.0021	0.0025	0.0028	0.003	0.0033	0.0036	0.004
153	0.0023	0.003	0.0035	0.004	0.0044	0.0047	0.005	0.0057
100	0.0036	0.0045	0.0055	0.0063	0.0069	0.0077	0.0082	0.0089

CONDITIONS: A 4-in. journal of steel 6 in. long fitted on its upper side with a gun metal brass, or half box, embracing somewhat less than one-half the journal circumference—the chord of the arc contact equal to 3.92 in.—and when running kept at a constant temperature 90° F.

TABLE 16.—COEFFICIENTS OF FRICTION FOR BATH OF OLIVE OIL—TEMPERATURE CONSTANT (*Tower*).

Nominal load lb. per sq. in.	Speeds							
	100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min	450 rev. 471 ft. per min.
Lb.								
573	.....	0.00102	0.00108	0.00118	0.00126	0.00132	0.00139	.....
520	.....	0.00055	0.00105	0.00115	0.00125	0.00133	0.00142	0.00148
415	.....	0.00093	0.00107	0.00119	0.0013	0.00140	0.00149	0.00158
363	.....	0.00084	0.0096	0.0011	0.00122	0.00134	0.00147	0.00155
258	0.00107	0.00139	0.00162	0.00178	0.00195	0.00213	0.00227	0.00243
153	0.00162	0.0020	0.00239	0.00267	0.003	0.00334	0.00367	0.00396
100	0.00277	0.00357	0.00423	0.00503	0.00576	0.00619	0.00663	0.00714

CONDITIONS: A 4-in journal of steel 6 in. long fitted on its upper side with a gun metal brass, or half box, embracing somewhat less than one-half the journal circumference—the chord of the arc contact equal to 3.92 in.—and when running kept at a constant temperature of 90° F.

TABLE 17.—COEFFICIENTS OF FRICTION FOR A BATH OF RAPE OIL—TEMPERATURE CONSTANT (*Tower*).

Nominal load lb. per sq. in.	Speeds								
	Actual load lb. per sq. in.	Temperature Fahr.	100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min.
328	582	90°	.....	.....	0.0107	0.0102	0.0098	.....	.....
310	551	82°	.....	0.0099	0.0099	0.0092	0.0099	.....	.....
293	520	76°	.....	0.0105	0.0105	0.0097	0.0097	.....	.....
275	498	77°	.....	0.0091	0.0091	0.0095	0.0103	.....	.....
258	458	78°	.....	0.0112	0.0095	0.0088	0.0084	0.0082	0.0083
205	364	82°	.....	0.0105	0.0087	0.0085	0.0078	0.0085	0.01
153	272	74°	0.0102	0.009	0.0096	0.0102	0.0105	0.0119	0.0125
100	178	75°	0.0105	0.0099	0.0109	0.0122	0.0133	0.0144	0.0154

CONDITIONS: A 4-in. journal of steel 6 in. long fitted on its upper side with a gun metal brass, or half box, embracing somewhat less than one-half the journal circumference—the chord of the arc of equal to contact of 2.25 in.—and when running kept at a constant temperature of 90° F.

TABLE 18.—COEFFICIENTS OF FRICTION OF RAPE OIL FROM A PAD UNDER THE JOURNAL (*Tower*).

Nominal load lb. per sq. in.	Actual load lb. per sq. in.	Speeds 100 rev. 105 ft. per min.	150 rev. 157 ft. per min.	200 rev. 209 ft. per min.	250 rev. 262 ft. per min.	300 rev. 314 ft. per min.	350 rev. 366 ft. per min.	400 rev. 419 ft. per min.
Lb.	Lb.							
258	317	.....	0.0056	0.0057	0.0063	0.0068	.....	.....
205	252	0.0132	0.0098	0.007	0.0077	0.0082	0.0087	.....
100	123	0.0144	0.0125	0.0146	0.0152	0.0163	0.0171	0.0178

CONDITIONS: Four-in. journal, 6 in. long; chord of arc-3 1/4 in.

TABLE 19.—COEFFICIENTS OF FRICTION FOR RAPE OIL FED BY SYPHON LUBRICATOR (*Tower*).

COEFEICIENT OF FRICTION OF COLLAR BEARINGS

The coefficient of friction for a collar bearing is much higher than for a journal bearing running under similar conditions. The reason lies in the difficulty with which the collar bearing is lubricated. There is little or no action tending to introduce oil between the surfaces as in journal bearings.

Load in lb. total	Speed in revolutions per minute 50	70	90	110	130
600	0.0450	0.0646	0.0433	0.0537	0.0642
1200	0.0375	0.0481	0.0496	0.0489	0.0475
1800	0.0357	0.0399	0.0361	0.0357	0.0371
2400	0.0286	0.0375	0.0361	0.0373	0.0410
2700	0.0354	0.0334	0.0346	0.0361	0.0378
3000	0.0347	0.0341	0.0348	0.0352	0.0356
3300	0.0337	0.0322	0.0348	.....	.....
3600	0.0312	0.0444	.....	.....	.....

CONDITIONS: Bearing faces 12 in. inside diameter, 14 in. outside diameter, one of soft steel, the other of gun metal, face grooved to distribute oil; lubrication sufficient to prevent seizing with a mineral oil.

TABLE 20.—COEFFICIENTS OF FRICTION FOR A COLLAR BEARING.

The coefficient of friction for collar bearing approaches that for journal bearings running at a low velocity and with poor lubrication, and tends to follow the laws for friction of solids rather than for fluids. It may be taken as from

0.04 to 0.05. Experimental data tend to show that the coefficient is independent of the speed of rubbing and decreases with an increase of load. At the same time this type of bearing is only usable for comparatively low unit pressures and for comparatively low speeds, unless the bearing surfaces are separated by a film of oil maintained by a pressure, or forced lubrication.

Experiments made by Mr. Tower and reported in the *Proc. Inst. M. E.*, 1888, page 179, were for a collar bearing 14 in. outside diameter, 12 in. inside diameter, having a soft steel face opposed to a gun metal face. Table 20 has been compiled from these experimental results.

FRICTION OF STEP BEARINGS

The friction of step bearings is about the same as of collar bearings. Step bearings are, in fact, no better than collar bearings except that they are generally of smaller diameters, hence have a lower speed and are therefore less liable to overheating. They present the same difficulties as regard lubrication.

It is very difficult to assign an average value of the coefficient for a step bearing. However, it may be taken as 0.01 for ordinary machine design.

In the *Proc. Inst. M. E.*, 1891, page 114, Tower gives results of experiments made by him with a flat pivot bearing having a steel step opposed to a manganese bronze-bearing surface. Table 21 gives some of the results of those experiments.

Load lb. per sq. in.	Speed in revolutions per minute									
	50		128		194		290		353	
	Oil drops per min.	Coeff. of fric.	Oil drops per min.	Coeff. of fric.	Oil drops per min.	Coeff. of fric.	Oil drops per min.	Coeff. of fric.	Oil drops per min.	Coeff. of fric.
20	20	0.0196	79	0.0080	196	0.0102	Con-	0.0178	Con-	0.0167
40	26	0.0147	82	0.0054	stream	0.0061	tinuous	0.0107	tinuous	0.0096
60	40	0.0167	80	0.0053	200	0.0051	stream	0.0078	stream	0.0073
80	50	0.0181	83	0.0063	200	0.0045		0.0064		0.0063
100	54	0.0219	98	0.0077	200	0.0044		0.0056		0.0057
120	56	0.0221	84	0.0083	168	0.0052		0.0048		0.0053
140	.....	.....	90	0.0093	158	0.0062		0.0046		.00053
160	.....	.....	84	0.0113	168	0.0068	200	0.0044		0.0054
260	.....	.....	.....	seized						
300	.....	.....	.....	seized						

CONDITIONS: Steel footstep running with a manganese bronze bearing diameter 3 in.; lubrication with a mineral oil as indicated.

TABLE 21.—COEFFICIENTS OF FRICTION OF PIVOT STEP BEARING.

Another series of experiments was made with white metal in place of the bronze bearing; the coefficient of friction in this case was a trifle greater than in the former, but the difference was so small that the results may be looked upon as practically identical. This is to be expected when the friction is the friction of the lubricant and not of the metallic surfaces.

COEFFICIENT OF LATERAL FRICTION

The endwise floating of a level shaft, provided there is end play between the bearings and journal shoulders, is something that is frequently noticed. In the construction of electric motors and generators it is very common to leave considerable end play as this lateral motion of the shaft aids in distributing oil throughout the bearings, and thus improves the quality of the lubrication. Anyone who has had occasion to touch a shaft under such conditions has probably been surprised at the very small amount of pressure that is required to make the shaft move; in fact, the change in the level of the floor by shifting of the weight of a person walking around the machine is often sufficient to make the shaft move endwise. This motion can be likened to that of a fine pitch screw rotating and moving in a fixed nut, with the mating thread formed by the oil film.

Professor Sweet in the *American Machinist* for February 16, 1911, page 315, gives this explanation for the smallness of the force required to move a leveled and well lubricated shaft in a lateral direction. "The Straight Line engines were, before the direct-connected age, built with a quarter of an inch end play in the main bearings and crank. When running, say at 250 revolutions, with 2-ton flywheels and 4 3/4-in. shafts, it was easy to slide the wheels and shafts back and fourth with a match. My explanation was this: In making, say three revolutions, the journals were sliding about 4 ft., and the match moved them 1/4 in. Thus, what the match has to do was to make up the difference between the base and the hypotenuse, where the base was 4 ft. and the perpendicular 1/4 in., which was very little, and as small as the match was, it was equal to do that little work."

John Goodman, in a paper in the *Proc. Inst. C. E.*, Vol. LXXXIX, page 424, discusses this question of lateral friction in an experimental apparatus, where a bronze half box composed of an alloy of 1 part of tin to 7 parts of copper, having a diameter of 2 in. and a length of 4 in., was run in contact with a manganese steel shaft. At a circumferential velocity of 200 ft. per minute, and a lateral velocity of 0.05 in. per minute, the lateral coefficient of friction was found to be 0.000021. This factor is given here more as a matter of interest than because of any practical value.

In general the lateral coefficient of friction varies directly as the lateral velocity, and inversely as the circumferential velocity.

COEFFICIENT OF FRICTION FOR SURFACES SLIDING IN STRAIGHT PATHS

The coefficient of friction for surfaces sliding in straight paths is not of much importance, as ordinarily guides, ways and V's of machines are not subjected to great pressures and do not move with high velocities. The condition of lubrication is that of unlubricated or very poorly lubricated surfaces. For metal in contact with metal under such conditions a safe factor of design for coefficient of friction is 0.3. For a more lengthy discussion of the subject reference should be had to Section I, where the experimental results of General Morin are presented.

John Aspinwall in *Proc. Inst. C. E.*, Vol. CXXXIII, page 13, gives results of a long series of experiments to determine the coefficient of friction of locomotive slide valves. The factors are: For the ordinary *D*-valve with its face vertical, 0.068; for a *D*-phosphor bronze partially balanced valve in a horizontal position, 0.0919; for an unbalanced cast-iron valve, 0.0878.

SECTION III

MATERIALS FOR BEARINGS

Probably nothing in connection with bearings has had a wider discussion or a greater amount of experimental investigation than the materials from which they are made. All of this implies that it is a question of importance; yet its degree of importance is determined to a great extent by the nature and quality of the lubrication that the bearings will receive in use. The first law of friction for perfectly lubricated surfaces, page 19, is that the coefficient of friction for well lubricated surfaces is practically independent of the nature of those surfaces. Thus, if the bearing surfaces can be kept separated by a film of oil the wear is the wear of the oil particles and not of the surfaces of the bearing and its journal. The principal requisites for the materials are that they should have sufficient mechanical strength to sustain the pressures to which they are subjected, and that they should tend to remain smooth and to conform to each other under the rubbing action, thus maintaining the conditions necessary for the formation of the oil film.

But the builder of machinery has no control over the lubrication that the bearings of his machine will receive after they have been put into service, and it is well known they are often abused and neglected. Thus the conditions of ordinary lubrication fall into the class of poorly lubricated surfaces and the question of what kind of material to use assumes considerable importance. Metals are, of course, the common materials; although wood and fiber have limited applications.

C. B. Dudley, late chemist to the Pennsylvania Railroad, states that a good bearing metal should have five characteristics. See *The Journal of the Franklin Institute*, February, 1892, page 83. These are:

1. The metal must have strength enough to sustain its load.
2. It must not heat rapidly.
3. It must work well in the foundry.
4. It must show a small coefficient of friction.
5. It must give a large amount of service with a small loss of metal by wear.

To these five we might add a sixth: In general it should be dissimilar from the metal of the journal with which it is to run. This is broadly true, except for hardened steel in contact with hardened steel, and cast iron in contact with cast iron. There seems to be no entirely satisfactory reason for this dissimilarity except it is claimed that wear is more rapid and friction greater when the

journal and its bearing are made from the same metal. It is possible that adhesion has something to do with it.

On page 5 it is pointed out that two metallic surfaces in intimate contact may be held together by a considerable force that we call adhesion or molecular attraction.

The ordinary combinations of metals used for spindles or shafts and their bearings are as follows:

Hard steel journals in contact with hard steel bearings, used for high speeds and moderate pressures.

It calls for excellent workmanship, as both journal and bearing must be accurately ground and carefully alined. Under such conditions the service rendered is good.

Hard steel journals with soft steel or wrought iron bearings. This combination is found in some presses where a hardened steel toggle pin fits into soft steel seats. It is generally considered to give good service for high pressures and low speeds.

Hard steel journals with bronze bearings. This is a very common combination in machine tools.

Hard steel journals with cast iron bearings. This combination is found to a limited extent in machine tools. The use of cast iron as a bearing metal has had a peculiar history, with strong advocates and just as strong opponents. It is of such importance that is is treated of in detail on page 43.

Soft steel journals with bronze bearings. This combination is the most commonly used of any among machine tools and high-grade machinery and gives good results.

Soft steel journals with babbitt bearings. This combination is the one most commonly used in machinery in general.

Soft steel journals with cast iron bearings. This combination is used to some extent in machine tools and to a much greater extent in medium and light weight machinery where the bearing pressures are small and the speeds low.

Soft steel journals and lignum-vitæ bearings. This combination has been used for line shaft bearings in the past. Now it is used for the steps of vertical water turbines. Tail shaft bearings of steamships are a bronze sleeve against lignum vitæ.

Cast iron with cast iron. This combination is a great exception to the disuse of like metals. It is employed for the step bearings of vertical steam turbines, to a limited extent for spindles and bearings of machine tools, is commonly used for the piston rings and cylinders of all kinds of reciprocating engines, has been used in the crossheads and slides of steam engines and air compressers, and is the combination commonly found in rectilinear slides and guides of all kinds of machinery.

WEAR OF BEARING METALS

The wear of bearing metals is of importance in railroad practice for it represents an enormous loss of valuable metal each year in the amount rubbed off and lost. In electrical machinery, such as motors and generators, the wearing of the bearing tends to decenter the armatures and rotors and thereby unequalize the air gap to the disadvantage of the operating of the machine. In any class of machinery where the alinement of the shafts is of importance, the wear of the bearing is likewise of importance, for this determines the position of the shafts.

Dr. Dudley in the *Journal of the Franklin Institute*, March, 1892, page 169, lays down three elements in the wear of bearing metals. These are:

1. That metal which will suffer the most distortion without rupture will wear best. Or, stated a little differently: That metal which has the greatest percentage of elongation will wear the best.

2. With a satisfactory elongation an increase in tensile strength will add to the wearing property of the metal.

3. With equal percentages of elongation and equal tensile strengths the finer the granular structure of the metal the longer will it wear.

The idea of wear is the rubbing off or rupturing of minute particles. Thus a metal which has the greatest tensile strength to resist rupture and the property of withstanding the greatest amount of distortion without rupture and a fine granular structure, so that the particles that are rubbed off are minute, can be considered as having the greatest resistance to wear. There are few experimental data to support the relationship between these variables, although the conclusion seems to be well founded that an increase of tensile strength at the expense of elongation is decidedly to the detriment of the wearing qualities.

In following paragraphs dealing with bronzes this question of wear will be again considered in quantitative factors.

CAST IRON

It is emphasized by its advocates and conceded by its opponents that cast iron under certain conditions will take on a hard glaze that makes a splendid wearing surface. This glaze is so hard that it will resist the cutting edge of a scraper. Furthermore, cast iron is porous and readily absorbs oil, thus to a degree becoming self-lubricating. It is granular in structure, in wear it rubs off in minute particles. It can be obtained in varying degrees of hardness, even chilled. Other advantages are the ease with which it is handled in the foundry and machine shop, and its cheapness.

The objection to it is summed up by saying that it is a "treacherous" metal. After glazing the bearing is very permanent and has great resistance to cutting;

but before glazing, if cutting is once started it takes but a very short time to ruin the bearing. With an assurance of adequate lubrication, the case against cast iron as a bearing metal would at once break down.

Professor Sweet has long been an earnest advocate of the cast-iron box, as frequent articles and discussions in the *American Machinist* show. William Sellers & Company, of Philadelphia, has been furnishing cast-iron lineshaft bearings for years.

R. K. LeBlond in the *American Machinist*, March 23, 1911, page 537, gives experiences with four experimental lathes fitted with spindles and bearings as follows:

1. Hardened steel spindle with cast-iron boxes.
2. Soft steel spindle with babbitt boxes.
3. Hardened steel spindle with bronze boxes.
4. Soft steel spindle with bronze boxes.

The soft steel spindles were of 60-carbon crucible steel; the bronze was made to the specifications of the Pennsylvania Railroad Company. After the end of some 8 years' service and treatment as far as possible identical for all four lathes, it was found that their rating as regards absence of wear and general satisfaction was in the order as given above; that is, the hardened steel spindle with cast-iron boxes was the best combination. Both spindle and boxes were in as good condition as when placed in the lathe, and from all appearances and tests showed absolutely no wear.

Mr. LeBlond says further: "The dry bronze box with soft spindle will stand more abuse than a dry cast-iron box and soft spindle; but a dry cast-iron box and hard spindle will withstand more abuse than the bronze." And again: "The hardened steel spindle and cast-iron box will stand as much neglect as any combination of metals, has a much longer life, will retain its accuracy for an indefinite period, will stand intermittent cuts or series of blows which would peen out and loosen babbitt, and, from our experience, is positively the best and most lasting bearing ever put into a machine tool."

The experience of the John Steptoe Shaper Company is shown by the following quotations taken from an article of Professor Sweet's printed in the *American Machinist.*

"We have been using cast-iron bearings on all our machines since July, 1909. Since that time we have turned out over 500 machines, and we have yet to receive the first complaint with regard to the bearings running hot; but we have made proper provision for the distribution of the oil over the bearings. We have chased spiral oil grooves in the shafts and have provided the bearings with ring oilers so that they are constantly flooded with oil. We believe that the cast-iron box is the best bearing that we have ever put in our machine."

Referring to Professor Sweet's own work, he says: "In the artisan lathe I

put a ground steel shaft in a solid cast-iron box. The outside of the box fitted in a conical piece with a nut at each end, but was not split, so when they first became loose the boxes will compress enough without splitting, and when not enough they can be split."

Cast iron bearings have been extensively used in shoe machinery for many years, and in textile machinery. One of the most troublesome places in the machine shop is in countershaft bearings. It has long been accepted practice in many places to bush countershaft hangers and loose pulleys with cast iron when the original bearings wore out.

Coleman Sellers, Jr., writes of the practice of his firm in regard to the use of cast-iron bearings and spindles as follows:

"William Sellers & Company have always believed in cast-iron bearings where the conditions of feed and pressure are not excessive and where efficient lubrication can be insured. Under such conditions we believe that a film of oil separates the metallic surfaces so that they do not come in contact. Where the load or speed are such that the film of oil is likely to be disturbed then it becomes necessary to use some material in the bearing not so likely to adhere or 'cut.' It has always been our practice to use cast-iron boxes in our own shafting, and we have had cases where boxes have run for 30 years, not only without cutting but without appreciable wear.

We have built many machines with cast-iron spindles and cast-iron bearings and with steel spindles in cast-iron bearings both with excellent results. We realize that such bearings must be thoroughly lubricated. They will not run well if dry. We have carried heavy vertical shafts on cast-iron steps with perfect results; but care has been taken to make these steps so large that the unit pressure would be low and the oil circulation was made very effective. Boring mill tables, planer tables, boring mill center spindles and boring mill steps are all emaples of cast-iron running on cast-iron, and it is well understood that under proper conditions of load, speed and lubrication such bearings are perfectly satisfactory."

In the *Proc. Inst. C. E.*, Vol. LXXXV, in a discussion by Dr. Goodman, it is stated that on the Brighton Railroad in England cast iron eccentric straps working on cast-iron hubs have run at least 100,000 miles without having been taken up.

These few quotations are sufficient to show the esteem in which cast iron is held as a bearing metal, provided it is used under advantageous conditions. The common prejudice against it is unfounded, but one must not lose sight of the fact that a cast iron bearing will not run dry.

SOFT ALLOYS OR WHITE METALS

Babbitts are white bearing metals whose most common constituents are tin, lead, and antimony. As distinguished from a brass or bronze, babbitt metal can be melted in an ordinary ladle. This is one of the governing reasons for its use as a lining for journal bearings. It is easily melted, easily

poured and run, easily anchored in place in its shell, has good anti-friction properties, and may be poured around the journal with which it is to run (although this is not approved practice). Thus the babbitting process is the easiest method known for obtaining well alined bearings.

B. R. Tompkins, writing in the Mechanical News for January, 1891, says: "For slow running journals, where the load is moderate, almost any metal that may be conveniently melted and will run free will answer the purpose. For wearing properties, with a moderate speed, there is probably nothing superior to pure zinc, but when not combined with some other metal it shrinks so much in cooling that it cannot be held firmly in the recess, and soon works loose; and it lacks those anti-friction properties which are necessary in order to stand high speed.

"For line shafting and all work where the speed is not over 300 or 400 r. p. m. an alloy of 8 parts zinc and 2 parts block tin will not only wear longer than any composition of its class, but will successfully resist a heavy load. The tin counteracts the shrinkage, so that the metal, if not overheated, will firmly adhere to the box until it is worn out. But this mixture does not possess sufficient anti-friction properties to warrant its use for fast running journals.

"Among all the soft metals in use there is none that possesses greater anti-friction properties than pure lead; but lead alone is impracticable, for it is so soft that it cannot be retained in the recess. But when by any process lead can be sufficiently hardened to be retained in the boxes without materially injuring its anti-friction properties, there is no metal that will wear longer in light fast running journals. With most of the best and most popular anti-friction metals in use and sold under the name of 'babbitt metal,' the basis is lead.

"Lead and antimony have the property of combining with each other in all proportions without impairing the anti-friction properties of either. The antimony hardens the lead, and when mixed in the proportion of 80 parts of lead by weight with 20 parts antimony, no other known composition of metals produces greater anti-friction or wearing properties, or will stand a higher speed without heat or abrasion. It runs free in its melted state, has no shrinkage and is better adapted to light high speed machinery than any other known metal. Care, however, should be manifested in using it, and it should never be heated beyond a temperature that would scorch a dry pine stick.

"Many different compositions are sold under the name of babbitt metal. Some are good, but more are worthless. But very little genuine babbitt metal is sold that is made strictly according to the original formula, most of the metals sold under that name are the refuse of type foundries and other smelting works, melted and cast into fancy ingots with special brands, and sold under the name of babbitt metal."

It is difficult to determine the exact formula used by the original discoverer,

Babbitt. Tin, copper and antimony were the ingredients, and from the best sources of information the original proportions in per cent. were as follows:

Tin	= 89.3	or 83.3	or 89.1.
Copper	= 3.6	or 8.3	or 3.7.
Antimony	= 7.1	or 8.3	or 7.4.

This metal, when carefully prepared, is probably one of the best metals in use for lining boxes that are subjected to a heavy weight and wear; but for light fast running journals the copper renders it more susceptible to friction.

BRONZE BEARING METALS

Next in importance to the babbitt is the bronze series of bearing metals. A bronze is commonly taken to mean an alloy of copper and tin with small amounts of other constituents. A brass is an alloy in which copper and zinc are the principal constituents.

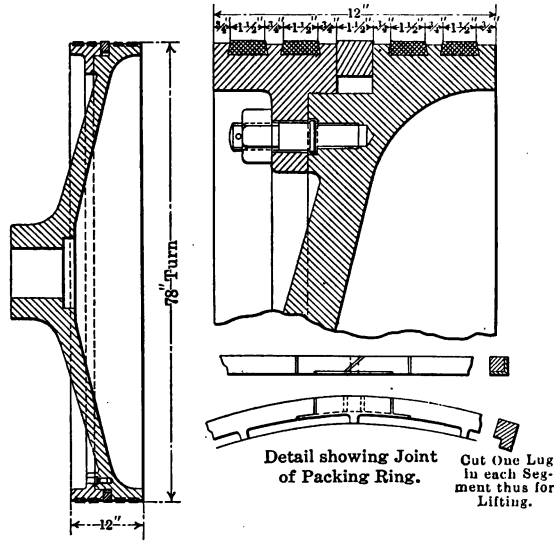


FIG. 7. BEARING METAL APPLIED TO A STEAM ENGINE PISTON.

In contrast to the softer metals, they must be melted in a crucible and cannot be manipulated in the easy manner in which the soft metal linings are made. As in the case of babbitt metals there are a multitude of bronzes on the market.

In an intermediate position between the babbitt series and the bronze series is a metal made by A. Allan & Son, known as Allan "red metal." It is composed of 50 per cent. copper and 50 per cent. lead. It is a soft metal, red in color, will withstand a temperature of 575° F. without injury, may be poured

into a lining shell, but not around a metal mandrel, and is extensively used for the bearing rings of reciprocating steam engine pistons. Fig. 7 shows the way in which it is used for this purpose. A similar metal is "Plumbic Bronze."

In the preceding paragraphs it was pointed out that lead is one of our most useful bearing metals when used with others to make an alloy. Two theories have been advanced as to the relative action of the constituents. The one is that the harder metal which forms a net work around particles of the softer metal is the important one. That is, it is the harder metal that takes the wear and determines the frictional value of the alloy; the softer metal merely fills up the spaces throughout the network of the harder matrix, and wears away more rapidly in order that the matrix may always be in contact with the journal.

The other theory is that the softer metal is the one that determines the anti-friction qualities of the alloy. This is the explanation that is more generally accepted, and in view of the fact that we know that lead is a most valuable constituent in these alloys, we are still further led to believe that it is the soft metal that is determinative.

Andrew Allan, of the firm of A. Allan & Son, was the first manufacturer of bronzes in America to point out the value of lead in large proportions as a valuable constituent of copper-lead-tin bearing metal. He has made such alloys since 1876. He has standard bronzes with different proportions of copper, lead and tin, for different services, as indicated below:

Number	Copper in per cent.	Lead in per cent.	Tin in per cent.	Uses
2	56	25	9	For severe service as mill pinion bearings.
4	59.5	35	5.5	For locomotive brasses.
5	50.5	45	4.5	For passenger and freight car brasses.
6	48.75	48.75	2.5	For service similar to that of hard babbitt.

Name of metal	Copper	Tin	Lead	Anti-mony	Zinc	Iron
Camelia metal	70.2	4.25	14.75	.....	10.2	0.55
Anti-friction metal	1.6	98.13	.....	.....	.....	trace
White metal	.....	.....	87.92	12.08	.....	.....
Car brass lining	.....	trace	84.87	15.1	.....	.....
Salgee metal	4.01	9.91	1.15	.....	85.57	.....

TABLE 22.—ANALYSES OF COMMON BEARING METALS (*Dudley*).

Name of metal	Copper	Tin	Lead	Anti-mony	Zinc	Iron
Graphite bearing metal	.....	14.38 Graphite—	67.73 none	16.73	.....	not determined
Antimonial lead	.....	.....	80.69	18.83	.....	.....
Carbon bronze	75.47	9.72 Carbon—	14.57 possible trace	.....	.....	.....
Cornish bronze	77.83	9.6 Phosphorus	12.4 —trace	.....	trace	trace
Delta metal	92.39	2.37	5.1	.....	.....	0.007
Magnolia metal	.....	.....	83.55	16.45	.....	.....
		Traces of iron, copper, zinc and possibly bismuth.				
American anti-friction metal	.....	.....	78.44	19.6	0.98	0.65
Tobin bronze	59	2.16	0.31	.....	38.4	0.11
Graney bronze	75.8	9.2	15.06	.....	.....	.....
Damascus bronze	76.41	10.6	12.52	.....	.....	.....
Manganese bronze	90.52	9.58 Manganese	..... —none	.....	.....	.....
Ajax metal	81.24	10.98 Phosphorus	7.27 or arsenic	0.37	.....	.....
American anti-friction metal	.....	.....	88.32	11.93	.....	.....
Harrington bronze	55.73	0.97	.....	.....	42.67	0.68
Carbox metal	.....	.....	84.33	14.38	trace	0.61
Hard lead	.....	.....	94.4	6.03	.....	.....
Phosphor bronze	79.17	10.22 Phosphorus	9.61 —0.94	.....	.....	.....
Ex. B. metal	76.8	8 Phosphorus	15 —0.2	.....	.....	.....

TABLE 22 (*Continued*).—ANALYSES OF COMMON BEARING METALS (*Dudley*)

ANALYSES AND PHYSICAL PROPERTIES OF BEARING METALS

Dr. Dudley, in *The Journal of the Franklin Institute*, February, 1892, page 87, gives a number of analyses of common bearing metals, mostly bronzes as determined in the laboratory of the Pennsylvania Railroad. Table 22 has been compiled from the results there given.

G. H. Clamer also in *The Journal of the Franklin Institute* gives a large number of analyses of bearing metals from which Tables 23–26 have been compiled. These are of value as showing a wide variety of both American and European practice.

Tin	Copper	Antimony	References
96	4	8	Quoted by Thurston. Used for ordinary bearings.
90	2	8	Quoted by Thurston. Quoted by Hiorns for bearings heavily loaded, used by Russian railroads for car bearings.
88.8	3.7	7.4	Quoted by Thurston and Bolley, as Karmarsch metal. Used in France in naval constructions.
87.5	12.5	.....	Quoted by Thurston, Karmarsch metal.
87	6	7	Quoted by Hiorns, for bearings heavily loaded.
85	5	10	Quoted by Ledebur and Hiorns as Jacoby metal for light pressure.
83.33	5.55	11.11	Used for car bearings, "Compagnies de l'Est, P. L. M., Ouest," etc.
83	6	11	Quoted by Ledebur. Used by Berlin railroads.
82	6	12	Quoted by Ledebur. Used by Orleans and the Western Austrian railroads.
82	8	10	Bearings for valve rods and eccentric collars, "Compagnie du Nord."
81	5	14	Quoted by Hiorns, for very hard bearings.
80	10	10	Quoted by Thurston. Used by Swiss railroads.
78.5	10	11.5	Quoted by Thurston. Used by Russian railroads.
76.7	7.8	15.5	Quoted by Ledebur and Thurston as English alloy.
76	7	17	Quoted by Hiorns, for bearings lightly loaded.
75	.....	25	Quoted by Thurston and Bolley, as Karmarsch metal.
73	9	18	Quoted by Thurston and Hiorns, for light pressures.
71.4	21.4	7.2	Quoted by Thurston. Karmarsch metal.
71	5	24	Thurston standard white metal. Used by the P. L. M. Company for packing of valves and eccentric collars.
67	22	11	Quoted by Thurston. Used by the Great Western Railway (England)
67	11	22	French state railroads.
33.3	22.2	44.5	Quoted by Hiorns. Dewrance metal for locomotives.
12	4	82	Quoted by Hiorns, for very hard bearings.

TABLE 23.—ALLOYS OF TIN, COPPER AND ANTIMONY (*Clamer*).

Lead	Tin	Antimony	References.
85	.....	15	Soft alloy quoted by Hiorns. Quoted also by Dudley.
84	.....	16	Quoted by Ledebur. For slow revolving pulleys.
80	12	8	Used by Eastern Railroad (France) for metallic packings.
77.7	5.9	16.8	Quoted by Thurston, as being the composition of Magnolia and Tandem metals.
76	14	10	Used for metallic packings by the Orleans and P. L. M. Railroads
73	12	15	Metallic packings of piston rods. Northern Company (France).
70	20	10	Metallic packings of eccentric collars. French state railroads.
68	15	17	Graphite (?) metal analyzed by Dudley.
60	20	20	Quoted by Ledebur and used for railroad bearings.
42	46	12	Quoted by Hiorns. Hoyles metal.
42	42	16	Quoted by Ledebur. Used for journal boxes, French state railroads.
37	38	25	Quoted by Thurston. Italian railroad companies.

TABLE 24.—ALLOYS OF LEAD, TIN AND ANTIMONY (*Clamer*).

Copper	Tin	Zinc	References
89	8	3	Quoted by Thurston for locomotive bearings.
89	9	2	Quoted by Thurston. Italian railroads.
87	8	5	Quoted by Bolley as a hard alloy.
86	14	.....	Quoted by Thurston for locomotive bearings.
84	14	2	Used by French State railroads for pieces subjected to alternative friction.
83	15	3	"Lafond" alloy for heavily loaded bearings.
82	18	.....	Quoted by Ledebur and Thurston. Car bearings of the "Compagnie du Nord."
82	15	3	Used by Orleans Railroad for valve-rod bearings.
82	16	2	Quoted by Ledebur for locomotive bearings. Used by French State railroads for pieces subjected to circular friction.
80	18	2	Quoted by Thurston and Bolley as "Lafond" alloy.
78.7	6.3	15	Quoted by Bolley for friction upon cast iron.
78	20	2	Quoted by Bolley and Thurston for car bearings.
77.4	15.6	7	Quoted by Haswell as a hard bronze for bearings.
58	28	14	Quoted by Thurston as "Margraff" alloy.
57	14	29	Quoted by Hiorns for bearings for propellor shafts.
56	28	16	Quoted by Thurston, "Fenton" alloy.
6	14	80	Quoted by Hiorns and Thurston. "Fenton" alloy for locomotive and car bearings.
5.5	17.5	77	Quoted by Ledebur, for high-speed horizontal shafts.

TABLE 25.—ALLOYS OF COPPER, TIN AND ZINC (*Clamer*).

Copper	Tin	Lead	Zinc	Iron	Anti-mony	Phos-phorus	References
10	.....	65	.....	.....	25	.....	Bearings for locomotives and tenders.
74	9.5	7	9.5	.....	.....	.....	Quoted by Thurston and Ledebur. Locomotive bearings.
79	8	8	5	.....	.....	.....	Quoted by Bolley and Ledebur. Bearings for engines.
8.3	7.6	3	83.3	.....	3.8	.....	Quoted by Ledebur as "Pierrot" metal. "Beugnot" white bronze used in France in naval constructions.
5.6	17.5	0.7	76.2	.....	.....	.....	White bronze used for ship engines.
1.6	46	.....	52	.....	0.4	.....	Dunnlevic and Jones metal.
.....	15	25	50	.....	.....	.....	Quoted by Ledebur, "Kniess" metal.
5	32	60	.....	.....	3	.....	Used by the western railroads (France) for piston rods and eccentric packings.

TABLE 26.—MISCELLANEOUS ALLOYS (*Clamer*).

An increasing amount of attention is being given to the physical properties of bearing metals; the tensile and compressive strengths, elongation, reduction of area and Brinell hardness number. It is recognized that a suitable chemical analysis alone is not sufficient to ensure physical characteristics that will give a good alloy in service. From the records of tests in the laboratory of the Pennsyl-

vania Railroad Table 27 has been arranged. This gives the physical properties of many commercial alloys.

Table 28 give representative chemical analyses of many of these same alloys, but the two tables must not be connected directly. For small variations in the constituents of such metals are common, while the effect of such variations cannot be stated definitely it is possible that small amounts of some elements may have a very decided influence upon the physical properties of the alloy that they enter.

This has led some firms to insist that nothing but ingot metal of known composition shall be used either in melting babbitts or in charging crucibles for bronze mixtures. In no case can scrap metal or the fins, or sprues, or defective castings from previous pourings be remelted. All such metal is sent to a refining room where it is analyzed and used to form the standard mixtures that are issued in the form of ingots to the babbitting room and bronze foundry. This case is exercised to keep the mixtures uniform, standard and under control.

In preparing the ingot metal, chemical tests alone are not relied upon, however, but the physical properties are likewise investigated. Considerable weight is given to the Brinell hardness number, although in some place the sclerescope hardness number is the one found and considered instead. In this connection it is interesting to note the same bronzes will give a sclerescope hardness number as high as high-grade alloy steel. This is probably due to the elasticity of the metals, for the sclerescope measures the recuperative power of the specimen tested when a permanent deformation is caused by the impact of the hammer. Sclerescope readings for different metals should not be compared.

Comparisons of Brinell and sclerescope hardness numbers for the same specimens indicate that there is some relationship between them, though just what has not been accurately determined.

In testing babbitts for hardness, and in fact other physical properties, care must be exercised to use specimens of the same general size and cast under similar conditions. Babbitt poured in a very thin section against a metal shell will tend to be harder than if poured in a thicker section.

Turning again to Table 27, two striking features are: first, the wide variations of the hardness numbers, and second, the increase in hardness after a compress in of 1/8 inch of the specimen. White metals run from 18 to about 35 Brinell hardness number, the highest being for Souther babbitt. The bronzes run from 40 to 106 Brinell hardness number, the latter being for a maganese bronze. This seems to be an unusually high number for a second determination only gives 68. This latter is closely approached by two of the Cramp's mixtures and exceeded by two other bronzes.

Plumbic bronze, which is a half-and-half mixture of lead and copper, has a Brinell hardness number of 21.8, thus being in the range of the babbitts.

Kind of material	Date	Tensile test			Brinell test 500 kg. load 10 mm. ball		Compression test	
					Hardness number		Original section	
		Tensile strength lb. per sq. in.	Elongation per cent. in 2″	Per cent. reduction of area	Natural	1/8″ comp.	lb. per sq. in.	per cent.
Plastic bronze	5– 5–06	20,430	11.5	8.6	45.0	55.6	68,786	46.1
Plastic bronze, 1 per cent. nickel	5– 5–06	21,340	10.1	8.2	45.0	53.0	64,490	44.2
Plastic bronze, 2 per cent. nickel	5– 5–06	19,340	2.8	2.3	45.0	57.6	53,390	34.6
Plastic bronze, 3 per cent. nickel	5– 5–06	23,670	4.0	3.3	49.0	61.6	57,230	33.6
Plumbic bronze	8– 9–06	.....	.....	.....	21.8	.....	.....	.....
Manganese bronze,	10–31–06	73,727	29.0	25.77	106.3	141.0	151,733	35.06
Manganese bronze	10–31–06	31,442	9.75	12.34	68.0	107.0	284,075	53.5
Phosphor bronze	1– 2–08	23,920	2.875 in 8″	.....	55.0	.....	.....	.....
Cyprus metal	1–2 –08	17,380	6.5 in 8″	.....	41.25	.....	.....	.....
Plastic bronze	1– 2–08	18,560	6.0 in 8″	.....	45.0	.....	.....	.....
Nickel bronze	1– 2–08	28,590	6.0 in 8″	4.41	65.00	73.0	107,600	36.0
Cramps hard bearing bronze	1– 8–08	31,780	2.0	13.04	65.0	80.0	213,400	59.5
Cramps special bronze	1– 8–08	27,875	15.0	16.39	40.0	50.0	100,600	37.8
Parsons white brass	1– 8–08	21,416	11.25	18.72	18.0	19.0	75,460	73.8
Vanadium bronze	2–12–10	19,985	2.5	1.2	78.0	104.0	262,050	61.0
Demo bronze	1–28–09	18,170	3.0	0.34	52.0	59.0	56,220	30.4
Acid proof copper	2–25–10	32,598	38.5	.....	76.0	.....	.....	.....
Hardened copper	2–25–10	34,069	31.0	.....	63.0	.....	.....	.....
Standard babbitt	10–11–10	11,780	5.0	3.14	27.7	29.0	29,150	59.0
Babbitt, German formula	10–11–10	8,726	.....	.....	21.1	22.5	33,895	59.2
Diamond S M. M. white bronze	10–11–10	11,710	4.0	5.76	25.1	31.5	36,755	54.8
Howitt metal	10–12–10	24,200	12.0	8.56	52.0	59.5	101,100	47.6
Dandelion metal (17811–1)	3–26–11	6,329	0.9	.....	30.7	23.9	17,360	41.4
Dandelion metal (17812–2)	3–26–11	7,247	0.6	.....	22.1	17.6	18,155	32.2
Dandelion metal (17813–3)	3–26–11	6,610	0.75	.....	29.1	24.7	17,672	27.9
Babbitt metal (Souther)	3–28–11	10,000	1.25	.....	34.8	48.0	28,410	44.1

TABLE 27.—PHYSICAL PROPERTIES OF VARIOUS BEARING METALS (*Pennsylvania Railroad Laboratory*).

	Composition in per cent.							
	Copper	Lead	Tin	Antimony	Nickel	Phosphorus	Sulphur	Zinc
Plastic bronze	64	30	5	.....	1	.....	.....	.....
Phosphor bronze	79.7	95	10	.....	.....	0.8	.....	.....
Cyprus bronze	64.75	30	5	.....	.....	.....	0.25	.....
Plumbic bronze	50	50	.....	.....	.....	.....	.....	.....
Parsons white brass	2.25	0.15	64.9	.....	.....	.....	.....	32.93
Demo bronze	60.67	32.97	4.6	.....	2.1	.....	.....	.....
Standard babbitt	3.7	.....	88.89	7.41	.....	.....	.....	.....
Shonberg M. M. metal	2.5	0.25	58.38	.....	.....	.....	.....	38.93
Souther babbitt	7	.....	84	9	.....	.....	.....	.....
German babbitt	5.55	.....	88.33	11.11	.....	.....	.....	.....

TABLE 28.—ANALYSIS OF BEARING METALS (*Pennsylvania Railroad Laboratory*).

The variation in babbitt metals purchased on brand only has been pointed out on page 46. Thus a careful purchaser will buy on specification, not on brand merely, and at the same time will watch the physical properties of the metals furnished.

A similar course will be followed by all who are careful about the bronzes used for bearings.

Name of metal	Composition in per cent.					
	Copper	Tin	Lead	Phosphorus	Arsenic	Relative wear in per cent.
Phosphor bronze	79.7	10	9.5	0.8	.....	100
Copper tin (1st experiment)	87.5	12.5	.....	.....	.....	148
Copper tin (2d experiment)	.....	same	.....	.....	.....	153
Copper tin (3d experiment.)	.....	same	.....	.....	.....	147
Arsenic bronze (1st experiment)	89.2	10	.....	.....	0.08	142
Arsenic bronze (2d experiment)	79.2	10	7	.....	0.08	115
Arsenic bronze (3d experiment)	79.7	10	9.5	.....	0.08	101
"K" bronze (1st experiment)	77	10.5	12.5	.....	.....	92
"K" bronze (2d experiment)	.....	same	.....	.....	.....	92.7
Alloy "B"	77	8	15	.....	.....	86.5

The physical properties of the phosphor bronze are tensile strength 30,000 pounds per square inche, elongation 6 per cent.

Physical properties of the alloy "B" are tensile strength 24,000 lb. per square inche, elongation 11 per cent.

TABLE 29.—RELATIVE WEAR OF VARIOUS BRONZES (*Dudley*).

WEAR OF CAR BRASSES

Dr. Dudley gives the wear of car brasses as a loss of 1 lb. of metal for every 18,000 to 25,000 miles run. From his experiments to determine the relative wear of different alloys in actual railroad service Table 29 has been arranged.

Referring to the last column it is seen that the relative wear in per cent. decreased steadily with an increase in the proportion of lead, and that the wear of the alloys containing no lead was far greater in every case than those containing that metal.

Copper	Tin	Lead	Wear in grams
85.76	14.90	.....	0.2800
90.67	9.45	.....	0.1768
95.01	4.95	.....	0.0776
90.82	4.62	4.82	0.0542
85.12	4.64	10.64	0.0380
81.27	5.17	14.14	0.0327
68.71	5.24	26.67	0.0204
64.34	4.70	31.22	0.0130

TABLE 30.—RELATIVE WEAR OF COPPER-TIN AND COPPER-TIN-LEAD ALLOYS (*Clamer*).

Copper	Tin	Lead	Zinc	Wear in grams
85.12	4.64	10.64	.....	0.0380
82.27	5.28	10.25	2.07	0.0415
79.84	4.71	10.30	5.44	0.0466
77.38	5.62	11.42	6.54	0.0472
74.28	4.68	10.61	11.04	0.0846

TABLE 31.—RELATIVE WEAR OF COPPER-TIN-LEAD-ZINC ALLOYS (*Clamer*).

Tables 30 and 31 are from Clamer's experiments. The first shows plainly the decrease of wear with the increase of the proportion of lead, while the second shows the increase of wear with the increase of zinc.

Recently compiled data of the wear of tender truck journal bearings are given in Table 32.

Kind	Size in.	Load in lbs. per sq. in. projected area	Wear per bearing in lbs. per 1000 miles	Kind of tender	Plain or filled bearings	Composition
Plastic bronze....	4 1/4x8	383	0.01343	.....	Plain	Plastic Bronze
Phosphor bronze..	4 1/4x8	383	0.01700	.....	Plain	Copper..... 64%
Plastic bronze....	4 1/4x8	383	0.01200	.....	Plain	Lead..... 30% Tin..... 5% Nickel..... 1%
Phosphor bronze..	4 1/4x8	383	0.01814	.....	Plain	Phosphor Bronze
Cyprus bronze....	4 1/4x8	383	0.01600	.....	Plain	Copper.....79.70%
Phosphor bronze..	4 1/4x8	383	0.02043	.....	Plain	Lead..... 9.50% Tin.....10.00% Phosphor..... 0.80%
Plastic bronze....	5 1/2x10	395	0.02159	5500 gall.	Plain	
Standard filled...	5 1/2x10	395	0.02348	5500 gall.	Filled	Cyprus Bronze
Standard filled...	5 1/2x10	395	0.02431	5500 gall.	Filled	Copper.....64.75% Lead.....30.00%
Lightened filled...	5 1/2x10	395	0.02437	5500 gall.	Filled	Tin..... 5.00% Sulphur..... 0.25%
Standard filled...	5 1/2x10	429	0.03542	7000 gall.	Filled	Filling Metal
Lightened filled...	5 1/2x10	429	0.03572	7000 gall.	Filled	Lead.....87% Antimony.....13%

4 1/2x8 inches used in both passenger and freight service.
5 1/2x10 inches 5500 gallon tenders—passenger service.
5 1/2x10 inches 7000 gallon tenders—freight service.

TABLE 32.—WEAR OF TENDER TRUCK JOURNAL BEARINGS
(*Pennsylvania Railroad Laboratory*).

REPRESENTATIVE PRACTICE IN THE USE OF BEARING METALS

The bearing metal known as the standard of the Bureau of Steam Engineering of the United States Navy, also called "anti-friction" or "anti-attrition" metal, has this composition:

Best refined copper	3.7 per cent.
Banca tin	88.8 per cent.
Regulus of antimony	7.5 per cent.

The percentages are by weight. The mixture must be well fluxed with borax and rosin in mixing.

The mixing of this anti-friction metal is a trick which must be learned. The best practice is to melt the copper, tin and antimony separately, adding the tin to the copper and the antimony to this mixture, fluxing it with borax with the proportion of about 1 1/2 lb. to 175 lb. of the mixture; but satisfactory

results are obtained by melting the copper first, dropping the cold tin into the melted copper and adding the antimony, which has been separately melted. This metal is carefully skimmed before pouring, and is poured into pigs and carried into stock as it stands.

The journal bronze used on battleships of the United States Navy has this composition: Copper 82 to 84, tin 12.5 to 14.5, zinc 2.5 to 4.5, iron (max.) 0.06, lead (max.) 1.00, all in per cent., with a normal of 83–13 1/2–3 1/2. It is used for bearings, bushings, sleeves, slides, guide gibs, wedges on water-tight doors and all parts subject to considerable wear.

The anti-friction metals commonly specified by the United States War Department are Parson's White Brass, Genuine Babbitt, Magnolia Metal, Phoenix Metal, and Shonberg's "M.M." Parson's White Brass from one analysis is composed of: tin 64.90, lead 0.15, copper 2.25, zinc 32.93; Shonberg M.M. white bronze: tin 38.38, zinc 38.93, copper 2.5, lead 0.25.

This comparison of the physical properties of three of these metals is taken from the *Journal of the American Society of the Naval Engineers*, Volume XIX.

	Naval white brass		Parson's white brass		Magnolia metal	
Original diameter, inches	0.75	0.75	0.759	0.758	0.751	0.748
height, inches.						
Elastic limit (about), lbs.....	8,000	8,000	5,000	5,000	5,000	4,500
Final compression loads, lbs..	36,000	37,000	35,000	36,800	37,000	36,500
Final height, inches.....	0.2	0.192	0.1805	0.18	0.161	0.166

The Westinghouse Electric & Manufacturing Company has developed most careful practice in preparing and handling bearing metals. In general, two kinds of babbitt are used; one having a tin base and the other a lead base. Not only are the proportions of the alloys carefully watched, but also the physical properties.

A standard test block to determine hardness is in the form of a disk 2–1/4 in. in diameter, 3/4 in. thick, and surfaced both sides. When tested by the Brinell method, the genuine babbitt shows a hardness number of about 25; similarly, the softer gives a number of about 21.

Care is used not to overheat the metals, for the harder the pouring temperature should not exceed 890° F.; the corresponding minimum temperature is about 660° F. In practice an effort is made to keep the pouring temperature about midway of there limits. For the softer metal the corresponding temperatures are 1000 and 840° F. This alloy is more easily abused by overheating than the harder one.

In general the softer metal is used for the regular run of work and the harder in railway motor bearings and bearings subjected to pound. Experience has

shown that for conditions of nearly steady pressure the softer metal is in many instances preferable to the harder one, being less liable to overheating in service.

Albert E. Guy gives the composition of the high-speed babbitt used in De Laval steam turbines as: copper 10, tin 80, and antimony 10 per cent. For low speeds the metal used is: lead 77, tin 6 and antimony 17 per cent.

RAILROAD PRACTICE

Representative practice of the Pennsylvania Railroad Company is as follows: For all axle bearings of passenger and freight cars, antimonial lead lining on ExB bronze shells; for locomotive driving wheel axle bearings, crank pin bearings, cross-head pin bearings, and small bearings generally, ExB, phosphor-, or plastic bronze; for cross-head shoes, block tin; for four-wheel engine truck boxes, standard babbitt strips set in bronze.

The specifications for journal bearing lining metal calls for a homogeneous alloy of lead and anitmony as free as possible from every other substance, and of the following composition: Lead 87 per cent., antimony 13 per cent. The rejection limits by analysis are: Lead less than 86 and more than 88 per cent., tin and antimony less than 12 and more than 14 per cent., tin more than 2 per cent., copper more than 0.5 per cent., or if the sum of the amounts of lead, tin, antimony and copper is less than 99.5 per cent.

The specifications for phosphor-bronze bearing metal call for a homogeneous alloy of copper, tin, lead, phosphorus, as free as possible from every other substance, and of the following composition: Copper 79.7 per cent., tin 10 per cent., lead 9.5 per cent., phosphorus 0.8 per cent. The rejection limits are as follows: Lots will not be accepted if the analysis as above described gives results outside the following limits: Tin, below 9 per cent. or over 11 per cent.; lead, below 8 per cent. or over 11 per cent.; phosphorus, below 0.7 or over 1 per cent.; nor if the metal contains a sum-total of any other substances than copper, tin, lead and phosphorus in greater quantity than 1 per cent.

The specification for standard babbitt calls for a homogeneous alloy of tin, antimony and copper, having the following composition in per cent.: tin 88.89, antimony 7.41, copper 3.7.

The ExB bronze analysis is:

Copper 76.75 per cent., limits	75.75 to 77.75
Lead 15 per cent., limits	13.5 to 16.5
Tin 8 per cent., limits	7 to 9
Phosphorus 0.25 per cent., limit	not below 0.20
Impurities, limit	not above 0.75

Robert Garbe, in *Die Danpflokomotiven der Gegenwart*, page 345, gives the composition of a white bearing metal that has been used successfully for 50 years by the Prussian State Railways; during that time it has remained un-

changed. It is used not only for the axle bearings of locomotives, but also in both freight and passenger cars.

It is made of copper, antimony and tin, in the following manner: 1 kg. (2.2 lb.) of copper is melted together with 2 kg. (4.4 lb.) antimony (regulus) and 6 kg. (13.2 lb.) of Strait's tin. The antimony is added when the copper is melted, and when both metals are fluid, the tin. This alloy is cast in thin plates and each 9 kg. (19.8 lb.) is remelted with 9 kg. (19.8 lb.) of pure tin. The whole is then cast in plates 15 mm. (0.6 in.) thick and is therewith ready for use. Larger quantities than those mentioned above ought not to be melted together at one time.

The individual constituents of the alloy must be as pure as possible. Thus the antimony should contain at the most 1 per cent. of impurities and among them not more than 0.1 per cent. of arsenic. Tin, on the other hand, should contain not more than 0.2 per. cent. of impurities. Furthermore, especially, lead and zinc ought not to be mixed in.

The constituents stated in per cent. are: tin 83.33, copper 5.55, antimony 11.11.

G. H. Clamer sums up American railroad practice in 1907 in the *Proc. A. S. T. M.*, Vol. VII, page 302. Continued experiments have shown that tin could be reduced and lead increased beyond the amounts given by Dr. Dudley, and that a satisfactory bearing metal could be made composed of 65 per cent. copper, 5 per cent. tin and 30 per cent. lead. This alloy is largely sold under the name of "plastic bronze." It has a compressing strength of about 15,000 lb. to the square inch and is found to operate without distortion in the bearings of the heaviest locomotives for the driving brasses, rod brasses and bushings, and for bearings of cars of 100,000 lb. capacity. Railroad specifications cover bearing alloys having a tin content of from 8 to 10 per cent. and lead from 10 to 15 per cent.

As railroad practice yields a large amount of old material, or scrap, which must be used, many car brasses are made from old metal. These contain copper 65 to 76 per cent., tin 2 to 8 per cent., lead 10 to 18 per cent., and zinc from 5 to 20 per cent. It is estimated that such bearings constitute from 50 to 75 per cent. of all the car bearings in use.

The Mesta Machine Company on rolling mill work uses two grades of babbitt and a bronze. For the general run of work, a lead babbitt is satisfactory having this composition in per cent., lead 75, tin 12.5, antimony 12.5. For high rubbing speeds a mixture is made of 1 part of the above and 2 parts of genuine babbitt. This genuine babbitt, alone, is used on rolling mill engines and in bearings subjected to shock and pound. Its composition in per cent. is: tin 82, copper 5.4, antimony 12.6. The bronze is a tough copper-tin-lead alloy very similar to Pennsylvania Railroad metal.

In the rolling mills for repairs any babbitt is considered satisfactory having about 10 per cent. tin, 12 to 14 per cent. antimony, and the rest unspecified. "Copper alloy" is also used for roll bearings.

AUTOMOBILE PRACTICE

The alloys division of the Standards Committee of the Society of Automobile Engineers in their report for June, 1911, specifies three bearing metals as follows:

BABBIT METAL, SPECIFICATION NO. 24

Tin	84 per cent.
Antimony	9 per cent.
Copper	7 per cent.

A variation of 1 per cent. either way will be permissible in the tin, and 0.5 per cent. either way will be permissible in the antimony and copper. The use of other than virgin metals is prohibited. No impurity will be permitted other than lead, and that not in excess of 0.25 per cent.

NOTE: This grade of babbitt is special, owing to the large amount of copper contained therein. It is used for the connecting-rod bearings of gasoline motor bearings, locomotive work, or for any service where machinery designers are confronted with severe operating conditions.

WHITE BRASS, SPECIFICATION NO. 25

Copper	3.00 to 6.00 per cent.
Tin, not less than	65.00 per cent.
Zinc	28.00 to 30.00 per cent.

Metal containing more than 0.25 per cent. impurities may be rejected.

NOTE: This alloy gives good results in automobile engines, but provision should be made to have it generously lubricated.

PHOSPHOR BRONZE BEARING METAL, SPECIFICATION NO. 26

Copper	80.00 per cent.
Tin	10.00 per cent.
Lead	10.00 per cent.
Phosphorus	0.05 to 0.25 per cent.

Impurities in excess of 0.25 per cent. will not be permitted.

NOTE: This is a metal similar to that specified by many railroads for various purposes. It is an excellent composition where good anti-frictional qualities are desired, standing up exceedingly well under heavy loads and severe usage. It should be used only against hardened steel in automobile construction.

RED BRASS, SPECIFICATION NO. 27

Copper	85.00 per cent.
Tin	5.00 per cent.
Lead	5.00 per cent.
Zinc	5.00 per cent.

A tolerance of 1 per cent. plus or minus will be allowed in the above percentages. Impurities in excess of 0.25 per cent. will not be permitted.

NOTE: A high grade of composition metal, and an excellent bearing where speed and pressure are not excessive. Largely used for light castings, and possesses good machining qualities.

ALLOYS FOR METALLIC PACKING

In metallic packing for piston rods, valve stems, and turbine shafts, we find a use of alloys similar to their use as bearing metals but with this important

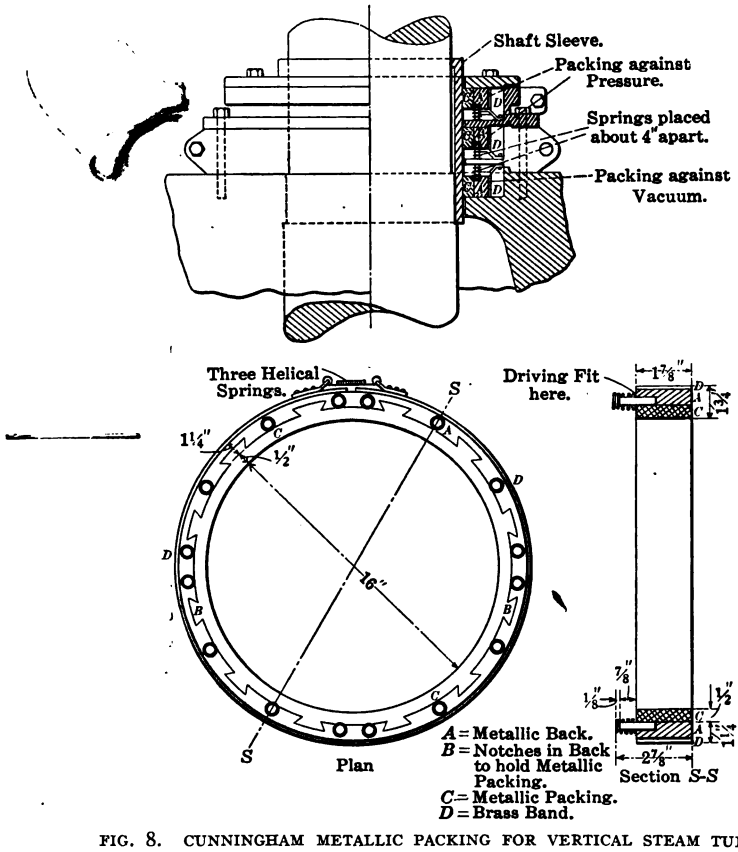


FIG. 8. CUNNINGHAM METALLIC PACKING FOR VERTICAL STEAM TURBINE.

difference: In packings the metal lining is not intended to carry any weight whatever but closes tightly around the shaft or rod and makes a steam-tight joint. However, as the rod and shaft are in motion, we have a condition of ordinary sliding friction, but differing again from bearing practice in that the parts are heated to a high temperature by steam.

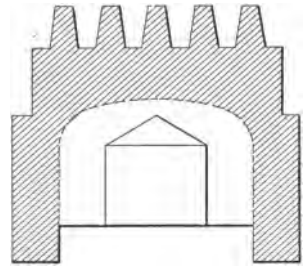


FIG. 9. SECTION OF LABYRINTH PACKING FOR STEAM TURBINE SHAFT.

Fig. 8 shows the Cunningham turbine shaft packing as used on Curtis verticle turbines in place of carbon packing. It consists of cast-iron quadrants filled with Allan "red metal," 50 per cent. lead and 50 per cent. copper, fitted together and surrounded by a brass band with helical tension springs.

The General Electric Company uses a similar packing but arranged in the form of a labyrinth, as shown in Fig. 9. Both of these packings are run without lubrication and are peculiarly well adapted for high-speeds and high temperatures.

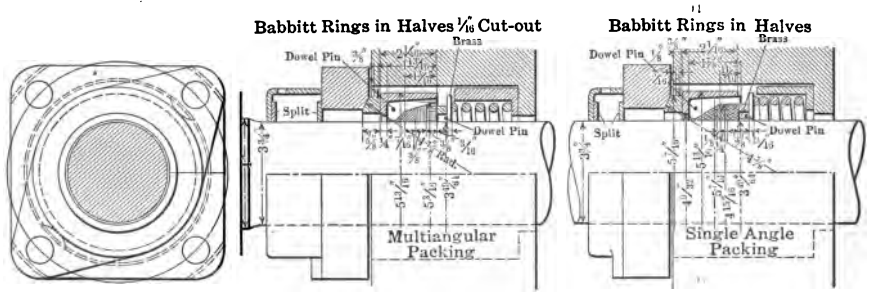


FIG. 10. ARRANGEMENT OF KING METALLIC PACKING FOR LOCOMOTIVE PISTON RODS.

To show the practice in regard to piston rods and valve stem packings, Fig. 10 is given. These are the "single angle" and "multi-angular" packing of the United States Metallic Packing Company, as applied in locomotive service. The babbitt mixture used for the single angle packing is 100 parts tin, 9 parts copper, and 6 parts antimony. For the multi-angular packing the

mixture is 83 1/3 parts lead, 8 1/3 parts tin, and 8 1/3 parts antimony. Allan "red metal" is also used.

In addition to these solid metal packings, there are many soft packings composed of metallic wool or fibers pressed together into a cord or rope and used in place of ordinary fiber packings. They are composed largely of lead with the addition of a small amount of tin or copper. A representative analysis is 88 lead and 12 copper.

VARIATIONS IN BEARING METALS

In the discussion of babbitt metals it is pointed out that many such "babbitts," placed on the market in fancy shaped ingots and under attractive names, are nothing but waste metals from type foundries and "clean up" from smelters. Of course, such metals vary in composition. If a particular mixture is desired for bearing purposes, it should be purchased from a reputable dealer according to a fixed analysis.

In the use of bronzes there is not only the variations in proportions of the constituent elements to be taken into consideration, but also the way in which the mixture is compounded. Under different foundry conditions the same constituent metals in identical proportions may give alloys with different physical properties. In adopting a bronze bearing metal one of the important points is to select an alloy that is easily handled under ordinary foundry conditions. If a precise mixture is desired, reputable metal dealers and foundrymen can be found who will guarantee to furnish either ingot metal or castings, in which none of the constituent metals will vary more than one half of 1 per cent. from the specification.

WOOD AS A BEARING MATERIAL

But little need be said in regard to the use of wood for bearings. In the early days of mill construction, lignum vitæ was successfully used for the bearings of mill shafting. This wood has a peculiar greasy character which, coupled with its extreme hardness and density, rendered it successful for moderate pressure and comparatively low speed.

Two other uses are deserving of mention: One is in the step bearings of water turbines where the step is constantly flooded with water. If the pressures and speed were not too high it has proved reasonably successful in this service.

The second, and at the present time practically its only application, is in the tail shaft bearings of steam ships and submerged guide bearings of water wheels. Here it is commonly put in in the form of staves, and as the bearing is open to the water it is constantly lubricated and cooled. Other kinds of wood have

been used to a limited extent, such as oak, wild pear, and apple tree wood. Several attempts have been made to put on the market wooden bushings soaked with a wax or grease to render it unnecessary to oil them. They are usually described as "oilless bearings." For very light pressures they have had a certain amount of success, although their use is by no means general.

GRAPHITE AS A BEARING MATERIAL

Graphite bearings usually consist of a brass or bronze shell having a series of openings or slots into which graphite in the form of a paste is forced under pressure and allowed to harden. Such bearings are intended to be run without oil, and are especially adapted for positions where oiling would be very difficult or under conditions of considerable heat. In operation the bronze shell supports the shaft and the graphite acts as a lubricant.

Another form of graphite bearing consists in casting a piece of netting having a number of slugs of graphite attached thereto into the bearing lining if of babbitt, or into the box or brass if of bronze. This type is not intended to run without lubrication, but the addition of the graphite in the form of these slugs equally spaced throughout the bearing surface is intended to reduce friction by adding a valuable solid lubricant.

MISCELLANEOUS MATERIALS

Hard fiber is used for small thrust collars and with low pressures and ample lubrication is satisfactory.

Knife-edge V's in weighing machines and computing scales are frequently made of agate with carefully polished surfaces.

The most frictionless bearings that are produced—if bearings they can be called—are mercury baths. They have been used in French lighthouses and to support certain parts of astronomical instruments. They permit of the freest motion of the floating members. This is an example of perfect fluid support. As a metal that will not amalgamate must be used for the float and casing, these parts are made of cast-iron. The specific gravity of mercury is 13.58.

SECTION IV

ALLOWABLE BEARING PRESSURES, SPEEDS AND TEMPERATURES

In Section II it is pointed out that values of the coefficient of friction for types of bearings and various kinds of lubrication are none too accurately known. The same condition exists to a certain extent for allowable bearing pressures. However, good practice is given in Table 33 for several kinds of machines and service. For light machinery the unit pressures are so small as to be negligible.

Kind of bearing and condition of operation	Allowable bearing pressure in pounds per square inch of projected area
Bearings for very low speeds and intermittent service as in turntables and bridges.	7000 to 9000
Railroad Practice	
Locomotive cross-head pin bearings	3000 to 4000
Locomotive crank pin bearings	1500 to 1700
Locomotive driving wheel journal bearings	... to 550[1]
Car axles bearings	300 to 325
Tender axle bearings	.... to 425
British Railway Practice	
Locomotive crank pin bearings	.... to 1400
Locomotive cross-head pin bearings	.... to 2000
Locomotive driving axle bearings	250 to 300
Car axle bearings	.... to 330

TABLE 32.—ALLOWABLE BEARING PRESSURES FOR MACHINERY BEARINGS.

[1] Pennsylvania Railroad Class E6 locomotives have a bearing pressure on bearings of main drivers of 556 lb. per square inch.

Kind of bearing and condition of operation	Allowable bearing pressure in pounds per square inch of projected area	
United States Naval Practice		
Main engine bearings	275 to 400	
Main engine crank pin bearings	400 to 500	
Steam turbine bearings	.... to 85	For weight alone.
Thrust bearings for torpedo boats	. .. to 50	
Merchant Marine Practice		
Main engine bearings	400 to 500	
Main engine crank pin bearings	400 to 500	
High-speed Stationary Engine Practice		
Main bearings	60 to 120	For dead load.
Main bearings	150 to 250	For steam load.
Crank pin bearings, overhung crank	900 to 1500	
Crank pin bearings, center crank	400 to 600	
Cross-head pin bearings	1000 to 1800	
Slow Speed Stationary Engine Practice		
Main bearings	80 to 140	For dead load.
Main bearings	200 to 400	For steam load.
Crank pin bearings	800 to 1300	
Cross-head pin bearings	1000 to 1500	
Air Compressor Practice [1]		
Straight line, steam driven, 100 lb. steam and air.		
Main bearings	160 to 237	
Crank pin bearings	565 to 700	
Cross-head pin bearings	628 to 820	

TABLE 33.—*Continued.*—ALLOWABLE BEARING PRESSURES FOR MACHINERY BEARINGS.

[1] Canadian Rand Company.

Kind of bearing and condition of operation	Allowable bearing pressure n pounds per square inch of projected area
Straight line, belt driven, center crank, 100 lb. steam and air.	
Main bearings	122 to 220
Crank pin bearings	244 to 402
Cross-head pin bearings	400 to 785
Straight line, belt driven, side crank, 100 lb. steam and air.	
Main bearings	178 to 227
Crank pin bearings	628 to 825
Cross-head pin bearings	628 to 825
Straight line, steam driven, side crank, 100 lb. steam and air.	
Main bearings	198 to 227
Crank pin bearings	462 to 825
Cross-head pin bearings	462 to 825
Duplex, Meyer cut-off, steam-driven, 100 lb. steam and air.	
Main bearings	157 to 200
Crank pin bearings	644 to 855
Cross-head pin bearings	850 to 1370
Duplex Corliss valve gear, steam driven, 100 lb. steam and air.	
Main bearings	115 to 141
Crank pin bearings	513 to 708
Cross-head pin bearings	732 to 1150
Direct-connected, motor driven main bearings	.... to 70
Gas Engine Practice	
Main bearings	500 to 700
Crank pin bearings	1500 to 1800
Cross-head pin bearings	1500 to 2000

TABLE 33.—*Continued.*—ALLOWABLE BEARING PRESSURES FOR MACHINERY BEARINGS.

Kind of bearing and condition of operation	Allowable bearing pressure in pounds per square inch of projected area	
Electrical Machinery Practice		
Generator and motor bearings	30 to 80	
Main engine bearings, driving generators	40 to 80	
Horizontal steam turbine bearings	400 to 60	
Vertical steam turbine steps	200 to 1000	
Rolling Mill Practice[1]		
		Rubbing velocity in feet per minute
Pinion housing bearings	30 to 50[2]	350 to 600
Roll housing bearings	100 to 2000[2]	350 to 600
Table roller bearings	30 to 50	150
Table line-shaft bearings	30 to 50	150
Main bearings of shears	1800 to 2500	50 to 65
Miscellaneous Practice		
Bearings for slow speed and intermittent load as in punch presses, shears, and the like.	3000 to 4000	
Main bearings of slow speed pumping engines	.... to 600	
Heavy line-shaft bearings, bronze or babbitt lined	100 to 150	
Light line-shaft bearings, cast-iron	15 to 25	
Heavy slow speed step bearings	.... to 2000	
Drill press thrust collars	.... to 325	
Angular-thrust bearing for boring mill tables	.... to 75[3]	

TABLE 33.—*Continued.*—ALLOWABLE BEARING PRESSURES FOR MACHINERY BEARINGS.

Tables 33A to 33F, inclusive, are from an article by G. W. Lewis and A. G. Kessler published the *American Machinist.* They give main bearing crank pin bearing, and wrist or piston pin bearing pressures for large stationary gas engines, both horizontal and vertical. Four maximum explosion pressures

[1] Mesta Machine Company, Pittsburg, Pa.

[2] These factors are of value as showing good practice, not for purposes of design. The diameters and lengths of the bearings are determined by the requirement of strength in the pinion and roll necks and their housings.

[3] Practice of Bullard Machine Tool Company.

in pounds per square inch of piston face are assumed, namely, 250, 300, 350, and 400 lb. These tables give a portion of the results of an extensive investigation of the designs of a large number of engines and thoroughly reflect modern practice.

D = cylinder diameter in inches.
Pm = maximum explosion pressure in pounds per square inch of piston face.
Dcp = bearing diameter of crank pin in inches.
Lcp = bearing length of crank pin in inches.
Kcp = maximum unit bearing pressure in pounds per square inch.
$Acp = Dcp \times Lcp$ in square inches.

$$\text{Max. Kcp} = \pi\frac{D^2}{4}\text{Pm} \div (\text{Dcp} \times \text{Lcp}) \quad \text{(A)}$$

D	4	8	12	16	20	Assumed
Dcp	1 1/2	3 1/8	4 3/4	6 3/8	8 1/8	
Lcp	1 5/8	3 1/4	4 7/8	6 9/16	8 3/8	
Acp	2.44	10.15	23.2	41.75	68	Dcp×Lcp=Acp
Pm			250			Assumed
Kcp	1290	1240	1220	1210	1150	From equation A
Pm			300			Assumed
Kep	1550	1485	1450	1450	1390	From equation A
Pm			350			Assumed
Kcp	1800	1730	1710	1690	1620	From equation A
Pm			400			Assumed
Kcp	2060	1980	1950	1930	1850	From equation A

TABLE 33A.—CRANK-PINS BEARING PRESSURES FOR HORIZONTAL STATIONARY GAS ENGINES.

D = cylinder diameter in inches.
Pm = maximum explosion pressure in pounds per square inch of piston face.
Dcp = bearing diameter of crank pin in inches.
Lcp = bearing length of crank pin in inches.
Kcp = maximum unit bearing pressure in pounds per square inch.
$Acp = Dcp \times Lcp$ in square inches.

D	4	8	12	16	20	Assumed
Dcp	1 5/8	3 1/4	4 7/8	6 1/2	8 3/16	
Lcp	1 5/8	3 5/8	5 5/8	7 5/8	9 5/8	
Acp	2.64	11.8	27.8	49.75	78.75	Dcp×Lcp
Pm			250			Assumed
Kcp	1190	1065	1035	1015	995	
Pm			300			Assumed
Kcp	1430	1280	1240	1215	1200	
Pm			350			Assumed
Kcp	1660	1490	1440	1420	1400	
Pm			400			Assumed
Kcp	1920	1720	1660	1620	1600	

TABLE 33 B.—CRANK-PIN BEARING PRESSURES FOR VERTICAL, STATIONARY GAS ENGINES.

D = cylinder diameter in inches.
Dmb = main bearing diameter in inches.
Lmb = main bearing length in inches.
Amb = projected area main bearing (one) = $Dmb \times Lmb$.
Pm = maximum explosion pressure in pounds per square inch of piston face.
Kmb = maximum unit bearing pressure in pounds per square inch considering explosion to occur on dead center.

D	4	8	12	16	20	Assumed
Dmb	1.3	3.1	4.85	6.6	8.4	
Lmb	2.6	6.75	10.8	14.9	19.1	
Amb	3.4	20.9	52.5	98.5	160.5	Dmb × Lmb
Pm			250			Assumed
Kmb	462	300	270	255	244	
Pm			300			Assumed
Kmb	553	360	324	307	293	
Pm			350			Assumed
Kmb	647	420	377	358	342	
Pm			400			Assumed
Kmb	738	503	430	408	391	

TABLE 33C.—MAIN BEARING PRESSURES FOR HORIZONTAL, STATIONARY GAS ENGINES.

D = cylinder diameter in inches
Dmb = bearing diameter of main bearing in inches
Lmb = bearing length of main bearing in inches
Amb = *Dmb*×*Lmb*
Pm = maximum explosion pressure in pounds per square inch of piston face.

D	4	8	12	16	20	Assumed
Dmb	1 1/2	3 1/2	5 1/2	7 1/2	9 1/2	
Lmb	3 1/2	6 3/4	10	13	16	
Amb	5.25	23.6	55	97.5	152	Dmb×Lmb
Pm			250			Assumed
Kmb	300	267	258	258	258	
Pm			300			Assumed
Kmb	359	320	310	310	310	
Pm			350			Assumed
Kmb	415	373	360	360	360	
Pm			400			Assumed
Kmb	481	414	414	414	414	

TABLE 33D.—MAIN BEARING PRESSURES FOR VERTICAL STATIONARY GAS ENGINES.

D = cylinder diameter in inches
Dwp = bearing diameter of piston pin in inches
Lwp = bearing length of piston pin in inches
Awp = projected area piston pin in square inches
Pm = maximum unit explosion pressure
Kwp = maximum unit bearing pressure in pounds per square inch
$Dwp = 0.0143\ D^2 + .7''$ (A)
$Lwp = 1.75\ Dwp$ (B)

D	4	8	12	16	20	
Dwp	.93	1.62	2.76	4.36	6.42	From Equation (A)
Lwp	1.6	2.8	4.77	7.52	11.15	From Equation (B)
Awp	1.49	4.54	13.2	32.8	71.5	Dwp×Lwp
Pm			250			Assumed
Kwp	2100	2760	2145	1530	1100	
Pm			300			Assumed
Kwp	2530	3320	2570	1840	1320	
Pm			350			Assumed
Kwp	2950	3880	3000	2150	1540	
Pm			400			Assumed
Kwp	3370	4425	3430	2455	1760	

TABLE 33E.—WRIST OR PISTON PIN BEARING PRESSURES FOR HORIZONTAL, STATIONARY GAS ENGINES.

D = cylinder diameter in inches
Dwp = bearing diameter of piston pin
Lwp = bearing length of piston pin
Awp = projected area piston pin in square inches
Kwp = maximum unit bearing pressure in pounds per square inch
Pm = Maximum unit explosion pressure.
$Dwp = 0.00795\ D^2 + 1\ 3/8''$ (C)
$Lwp = 1.82\ Dwp$ (D)

D	6	8	12	16	20	
Dwp	1 5/8	1 7/8	2 1/2	3 3/8	4 1/2	Equation (C)
Lwp	2 7/8	3 3/8	4 1/2	6 1/8	8 1/8	Equation (D)
Awp	4.67	6.33	11.25	20.65	36.6	Dwp×Lwp
Pm			250			Assumed
Kwp	1510	1990	2520	2430	2145	
Pm			300			Assumed
Kwp	1810	2380	3020	2920	2580	
Pm			350			Assumed
Kwp	2120	2780	3520	3410	3010	
Pm			400			Assumed
Kwp	2420	3190	4030	3900	3440	

TABLE 33F.—WRIST OR PISTON PIN BEARING PRESSURES FOR VERTICAL STATIONARY GAS ENGINES.

Tables 33G to 33L, inclusive, are from an article by G. W. Lewis and A. G. Kessler published in the *American Machinist.* They give the maximum unit pressures and rubbing speeds on the bearings of automobile engines

for various assumed explosion pressures and for various sizes of engine cylinders. They were developed from an investigation of some 30 American automobile engines and represent average practice.

D = cylinder diameter in inches,
Dcb = bearing diameter of center bearing in inches,
Lcb = bearing length of center bearing in inches,
Kcb = maximum unit bearing pressure on center bearing in pounds per square inch,
Pm = maximum explosion pressure in pounds per square inch of piston face.

$$Dcb = 0.32\ D + 0.3'' \quad (5)$$
$$Lcb = 2.8\ Dcb - 2.2'' \quad (6)$$

D	4	4 1/2	5	5 1/2	
Dcb	1 9/16	1 3/4	1 7/8	2 1/16	From equation (5).
Lcb	2 3/16	2 3/4	3 1/16	3 5/8	From equation (6).
Acb	3.42	4.3	5.75	7.5	Dcb × Lcb.
Pm			250		Assumed.
Kcb	690	615	620	575	
Pm			300		Assumed.
Kcb	830	730	750	690	
Pm			350		Assumed.
Kcb	970	855	870	810	
Pm				400	Assumed.
Kcb	1100	980	1000	920	

TABLE 33G.—MAXIMUM UNIT BEARING PRESSURE ON CENTER BEARINGS OF AUTOMOBILE ENGINES.

D = cylinder diameter in inches,
Dfb = bearing diameter of front bearing in inches,
Lfb = bearing length of front bearing in inches,
Kfb = maximum unit bearing pressure on front bearing in pounds per square inch,
Pm = Maximum explosion pressure in pounds per square inch of piston face.

$$Dfb = 0.32\,D + 0.3'' \quad (7)$$
$$Lfb = Dfb + 1\ 1/8'' \quad (8).$$

D	4	4 1/2	5	5 1/2	
Dfb	1 9/16	1 3/4	1 7/8	2 1/16	From equation (7).
Lfb	2 11/16	2 7/8	3	3 3/16	From equation (8).
Afb	4.2	5.02	5.63	6.6	Dfb × Lfb.
Pm			250		Assumed.
Kfb	560	595	640	670	
Pm			300		Assumed.
Kfb	665	710	775	800	
Pm			350		Assumed.
Kfb	780	830	900	930	
Pm			400		Assumed.
Kfb	890	945	1030	1075	

TABLE 33H.—MAXIMUM UNIT BEARING PRESSURE ON FRONT BEARINGS OF AUTOMOBILE ENGINES.

D = cylinder diameter in inches,
Drb = bearing diameter of rear bearing in inches,
Lrb = bearing length of rear bearing in inches,
Arb = projected bearing area of rear bearing in square inches,
Krb = maximum bearing pressure on rear bearing in pounds per square inch,
Pm = maximum explosion pressure in pounds per square inch of piston face.

$$Drb = 0.32D + 0.3'' \quad (9)$$

$$Lrb = 5.3\ Drb - 5.3'' \quad (10).$$

D	4	4 1/2	5	5 1/2	
Drb	1-9/16	1-3/4	1-7/8	2-1/16	From equation (9)
Lrb	3	4	4-5/8	5-5/8	From equation (10)
Arb	4.7	7	8.67	11.6	Drb × Lrb.
Pm			250		Assumed.
Krb	565	495	495	465	
Pm			300		Assumed.
Krb	660	575	575	535	
Pm			350		Assumed.
Krb	760	660	655	605	
Pm			400		Assumed.
Krb	855	735	735	675	

TABLE 33I.—MAXIMUM UNIT BEARING PRESSURE ON REAR BEARINGS OF AUTOMOBILE ENGINES.

D = cylinder diameter in inches,
Dwp = bearing diameter of wrist pin in inches,
Lwp = bearing length of wrist pin in inches,
Awp = projected bearing area of wrist pin in square inches,
Pm = maximum unit explosion pressure in pounds per square inch. of piston face.

$$Dwp = 0.34\,D - 0.53'' \quad (1)$$

$$Lwp = 2.25\,Dwp \quad (2)$$

Kwp = maximum unit bearing pressure in pounds per square inch.

$$Kwp = \frac{Pm\,D^2\,0.7854}{Awp} \quad (a)$$

D	4	4 1/2	5	5 1/2	
Dwp	7/8	1	1 3/16	1 3/8	From equation (1).
Lwp	2	2.25	2 5/8	3 1/16	From equation (2).
Awp	1.75	2.25	3.12	4.2	Dwp × Lwp.
Pm			250		Assumed.
Kwp	1800	1780	1570	1420	Equation (a).
Pm			300		Assumed.
Kwp	2150	2130	1890	1700	Equation (a).
Pm			350		Assumed.
Kwp	2510	2480	2200	1980	Equation (a).
Pm			400		Assumed.
Kwp	2870	2840	2510	2270	Equation (a).

TABLE 33J.—MAXIMUM UNIT WRIST PIN BEARING PRESSURES IN AUTOMOBILE ENGINES.

D = cylinder diameter in inches,
Dcp = bearing diameter of crank pin in inches,
Lcp = bearing length of crank pin in inches,
Acp = projected bearing area of crank pin in square inches,
Kcp = maximum unit bearing pressure on crank pin in pounds per square inch,
Pm = maximum unit explosion pressure in pounds per square inch of piston face.

$$Dcp = 0.32\,D + 0.3'' \qquad (3)$$

$$Lcp = 1.35\,Dcp \qquad (4)$$

$$Kcp = \frac{Pm\,D^2\,0.7854}{Acp} \qquad (b)$$

D	4	4-1/2	5	5-1/2	
Dcp	1 9/16	1 3/4	1 7/8	2 1/16	From equation (3)
Lcp	2 1/18	2 3/8	2 1/2	2 3/4	From equation (4)
Acp	3.32	4.17	4.68	5.68	Dcp × Lcp.
Pm			250		Assumed.
Kcp	945	960	1050	1050	From equation (b).
Pm			300		Assumed.
Kcp	1130	1150	1260	1260	From equation (b).
Pm			350		Assumed.
Kcp	1320	1350	1470	1470	From equation (b).
Pm			400		Assumed.
Kcp	1510	1540	1675	1675	From equation (b).

TABLE 33K.—MAXIMUM UNIT CRANK PIN BEARING PRESSURES FOR AUTOMOBILE ENGINES.

D = cylinder diameter in inches,
L = stroke in inches,
P_{mean} = mean total pressure on piston in pounds for entire cycle (assumed),
RPM = revolutions per minute at 1000 ft. per minute piston speed,
Km = mean unit bearing pressure on bearings in pounds per square inch,
V = rubbing speed in feet per second,
W = work of friction = $(Km \times V)$ in foot pounds per second,
L = 1.1 D.

D		4	4-1/2	5	5-1/2	Average
L		4-3/8	5	5-1/2	6	
R.P.M.		1370	1200	1090	1000	
Rubbing speed in ft. per min. on bearings listed below.		560	550	535	540	546
V		9.35	9.15	8.9	9	9.1
P_{mean}		252	320	395	475	
Crank-pin bearing.....	Km	76	77	84	84	80
	W	710	705	745	755	729
Center bearing	Km	55	49.2	49.8	46	50
	W	515	450	445	415	456
Front bearing.....	Km	44.7	47.5	52	53.4	49.4
	W	418	435	462	480	449
Rear bearing.....	Km	56	45.5	43.5	38	45.7
	W	523	415	388	342	417

TABLE 33L.—AVERAGE RUBBING SPEED AND WORK OF FRICTION FOR AUTOMOBILE ENGINES.—Computed for a piston speed of 1000 ft. per minute and a mean pressure of 20 lb. per square inch of piston face.

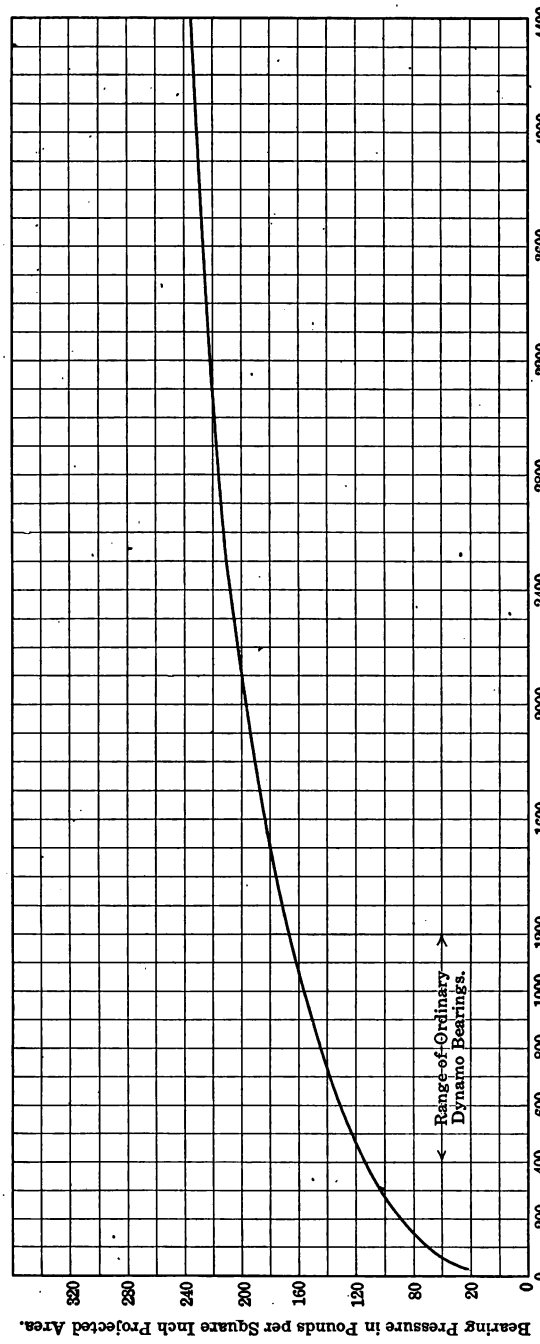


FIG. 11. RELATION BETWEEN RUBBING SPEED AND SAFE MAXIMUM PRESSURE—FOR PERFECT FILM LUBRICATION.

Fig. 11 gives maximum safe bearing pressures for rubbing speeds up to 5000 ft. per minute and for perfect film lubrication. It shows the practice of the General Electric Company for the bearings of electrical machinery and is representative of the best of American practice.

REPRESENTATIVE GERMAN PRACTICE

Representative German practice in regard to unit pressures for journal bearings as found on page 742 *Des Ingenieurs Taschenbuch, Hütte,* Vol. 1, is given below with the metric units transformed into corresponding English units in round numbers.

Materials in contact	Pressure in pounds per square inch of projected area.
Hardened crucible steel on hardened crucible steel	2130
Hardened crucible steel on bronze	1280
Unhardened crucible steel on bronze	850
Mild steel with a smooth compact surface on bronze	57
Mild steel with not an entirely clear surface on cast-iron or bronze	425
Mild steel with not an entirely clear surface on cast iron	355
Mild steel on lignum vitæ with water lubrication	355

In explanation of the above values, we read: "If the total pressure even when the journal is at rest maintains approximately of the same magnitude and direction (for example, in heavily loaded shafts, heavy gears, etc.) the unit pressure should be taken smaller than above.

"For journals (or bearings) which oscillate only the unit pressure may be considerably higher.

"For the bearings of rope and chain sheaves, etc., which turn intermittently and on which the wear is either small or unimportant, the above unit values, 850 to 355 lb., may be doubled or tripled.

"For crucible steel crank and cross-head pins running on bronze in ordinary steam engines the unit pressure may be taken from 710 to 995 and 1070 to 1130 lb., respectively; for similar purposes on locomotive 1420 and 2130 lb., respectively; on high speed steam engines 570 to 710 lb., respectively. For fly wheel bearings of steam engines use 213 to 217 lb.

"For crank pin bearings of punches and shears the unit pressure may be as high as 2845 lb. per square inch of projected area."

PERMISSIBLE VELOCITY OF RUBBING

There are no well defined limits for rubbing velocities. In many classes of machinery the speed is so low that the velocity is an unimportant factor. With

the exception of power generating machinery, particularly steam turbines, high speed is usually accompanied with very low pressure. For instance, in machine tools speeds are uniformly low with the exception of grinding machinery, and there the pressures are slight.

In *Power*, February, 1893, C. J. Field gives 350 ft. per minute as the maximum surface velocity for the main bearings of steam engines in electrical service with grease lubrication. The corresponding maximum pressure per square inch of projected area of journal is 80 pounds.

The ordinary range of velocities for electric motors and generators, as indicated by the curve in Fig. 11, is from 400 to 1200 ft. per minute.

Steam turbine practice in America is from 2400 to 6000 ft. per minute. However, this higher figure has been exceeded. One case that came to the author's attention was of a horizontal steam turbine with a bearing rubbing velocity of 8000 ft. per minute.

PERMISSIBLE RISE IN TEMPERATURE

A bearing shows distress by heating—by becoming a "hot box." Within limits there is no objection to the rise in temperature; in fact, it must take place. It is sometimes said that the hotter a bearing runs the hotter it can run. The reason for this statement lies in the fact that the higher the temperature the less the viscosity of the lubricant, therefore the less the work of friction and the less the amount of energy transformed into heat at a given speed.

With soft metal bearings the limiting temperature is one at which the lining will melt. With other materials the limiting temperature is one at which lubrication becomes insufficient and the bearing seizes.

In practice it is necessary to design bearings to run at a much lower temperature than will cause damage because of the requirements of the average customer. Such a maximum temperature is from 140 to 160° F. It is probably true that the average bearing running at that temperature could just as well run at a temperature of 200° F. Occasionally manufacturers take advantage of this fact in designing a machine to be located in some out-of-the-way place where it will not have close inspection, or will not be approached by the ordinary passer-by who might place his hand upon a bearing, and a high temperature is allowed.

There is no valid objection to running a bearing hot, provided the temperature reached is not sufficient to do harm and provided there is no progressive increase in heat indicating trouble. This is opposed to the ideas of some machinery users who look upon any bearing that warms up as improperly designed. There is no reason in expecting or demanding that a machinery bearing should run stone cold. In the very nature of the device, energy must be transformed into heat, and this heat will cause a rise in the temperature of

the bearing until a condition is reached where the heat radiated and dissipated in a unit of time balances the amount liberated in the same time.

In machine-tool practice low running temperatures are the rule. This indicates a very ample factor of safety. The Bullard Machine Tool Company has fixed the maximum running temperature under load for bearings of the machines that they build as 110° F. Every machine is tested in this particular before shipment. The actual running temperatures are considerably lower as shown by the following tabulation, giving the log of tests of six machines.

Machine		Running Temperature of Bearings in Degrees Fahrenheit							
		Speed box	Head-stock	Table pinion	Power traversing bracket		Feed works		Driving pulley bracket
					Upper	Lower	Upper	Lower	
24-inch vertical turret lathe.	1– 70	65	76	68	70	66	65	70
	2– 80	68	79	71	74	68	69	80
	3– 85	68	80	72	74	68	69	78
24-inch vertical turret lathe.	1– 79	61	.....	74	70	.....	.....	.....
	2– 95	65	.....	80	76	.....	.....	.....
	3– 91	67	.....	80	78	.....	.....	.....
24-inch vertical turret lathe	1– 74	64		77	76	.....	.....	.....
	2– 96	68	Rail	80	80	.....	.....	.....
	3– 95	68	raising bracket	79	78	.....	.....	.....
36-inch vertical turret lathe.	1– 75	62	73	68	66	.....	.....	.....
	2– 81	65	78	70	70	.....	.....	.....
	3– 80	65	80	71	72	.....	.....	.....
36-inch vertical turret lathe.	1– 82	70	78	86	88	.....	.....	.....
	2–100	72	82	84	85	.....	.....	.....
	3– 97	72	82	82	83	.....	.....	.....
36-inch vertical turret lathe.	1– 80	64	85	75	74	.....	.....	.....
	2– 85	65	110	80	78	.....	.....	.....
	3– 86	66	112[1]	80	79	.....	.....	.....

The three sets of readings in each case were taken: 1, at the end of the first hour's run *without load;* 2, at the end of the second hour's run *with load;* 3, at the end of the third hour's run *without load.*

[1] After correction of this bearing the temperature dropped to 78° at the end of a 2-hr. run

PRODUCT OF UNIT PRESSURE AND VELOCITY

In designing bearings neither the pressure nor the circumferential speed alone can serve as a standard for calculating dimensions. The product of the pressure by the speed and the coefficient of friction must be taken into account, that is, the work of friction.

In Section II it is shown that the product of the unit pressure and the coefficient of friction remains almost a constant for a wide range of pressures. This has been taken advantage of by many designers to establish a limiting factor of design commonly referred to as C, which represents the product of the pressure in pounds per square inch and the velocity in feet per minute.

However, the values of this factor differ so widely that no attempt will be made here to tabulate them for general types of bearings. To give an idea of the range, F. W. Taylor, *Trans. A. S. M. E.*, Vol. XXVII, gives C for mill work as 24,000 and for cast-iron bearings with ordinary lubrication as 12,000. The same factor used in turbo-generator design for forced lubrication and water cooling is 360,000. It is probable that these may be taken as limiting values, but it is impossible to give consistent data for other kinds of bearings in order to complete the series.

Tower mentions two cases in which his experimental bearing seized, the conditions being a bronze half box, a steel journal and bath lubrication. In one instance the bearing seized at a pressure of 573 lb. per square inch and a speed of 419 ft. per minute. This is equivalent to a factor of 240,000. In the other case the pressure was 625 lb. per square and the rubbing velocity 366 ft. per minute. The corresponding factor is 228,000.

One instance that has come to the author's attention is of a large turbo-generator running with a bearing pressure of 41 lb. per square inch and rubbing velocity of 8,000 ft. per minute. The value of C in this case is 328,000.

Some of the greatest values for this factor that have come to the author's attention are from some experiments made by the General Electric Company. In one case a bearing was run at a pressure of 285 lb. per square inch and a rubbing velocity of 2,700 ft. per minute—$C = 770{,}000$. In the second case the pressure was 200 lb. and the velocity 5,000 ft.—$C = 1{,}000{,}000$.

The Westinghouse Machine Company has carried this experimentally to 1,720,000; 400 lb. pressure, 4300 ft. per minute, and for long runs.

In spite of the fact that it is impossible to give values for this factor that seem to be consistent with themselves and in sufficient number to cover the ordinary range of commercial bearings, pressures, and speeds, the designer should not think it is of no value. On the contrary, it is of considerable value as a rough check upon design. It is wise for any one having to do with bearing design to tabulate this factor carefully for all important bearings coming within his experience, to arrive at limiting values beyond which he will know that the running of the bearing is liable to be unsatisfactory if not impossible.

SECTION V

DESIGN OF JOURNAL BEARINGS

The important points to consider in proportioning and designing journal bearings may be grouped under six heads:

a. Proportions and dimensions of diameter and length.

b. Selection of the bearing material.

c. Provisions for anchoring the bearing metal if a soft lining, and provisions for adjustment to preserve fit and alinement.

d. Clearance between journal and bearing.

e. Provisions for lubricating.

f. Provisions for radiating and dissipating heat.

RATIO OF BEARING LENGTH TO DIAMETER

In proportioning bearings, the designer must harmonize conflicting conditions. On the one hand the shaft must be sufficiently strong and rigid to sustain its load without undue deflection (this calls for large diameters); again the bearing pressures must not exceed a certain amount in order that an oil film may be established and maintained (this calls for large diameters and long bearings). On the other hand the circumferential speed must be kept as low as possible, thus keeping down the work of friction and reducing the amount of heat liberated and the risk of seizing (this calls for small diameters). Again the bearing length should be short, thus rendering the surfaces more effective and minimizing the danger of a local concentration of pressure from the deflection of the journal (this calls for short bearings).

In actual designing the total load to be carried by the bearing divided by the allowable unit pressure, gives at once the projected area of the bearing. The smallest permissible diameter of journal can then be found by determining the shaft deflection under running conditions. This investigation can best be made by graphic statics. A valuable discussion of this subject will be found in *Elements of Graphic Statics*, by Cathcart and Chaffee, at page 269. The diameter thus determined can be checked by finding out if its circumferential rubbing velocity is greater than allowable.

If the first proportions selected are found to be beyond the limits of good practice, another set can be taken and the investigation repeated until the

proportions are deemed satisfactory. Section IV gives allowable bearing pressures and velocities.

It is seen from the preceding principles that the diameter and length of a bearing should both be kept as small as possible, consistent with the necessary strength and stiffness; the same conclusion is reached from the viewpoint of economy of material.

Practice has tended to standardize the ratio of bearing length to diameter and the following tabulation gives representative proportions. Here l is the bearing length and d the diameter.

Type of bearing	Values of $\frac{l}{d}$
Marine engine main bearings	1 to 1.5
Marine engine crank pin bearings	1 to 1.5
Stationary engine main bearings	1.5 to 2.5
Stationary engine crank pin bearings	1
Stationary engine cross-head pin bearings	1 to 1.5
Ordinary heavy shafting with fixed bearings	2 to 3
Ordinary shafting with self-adjusting bearings	3 to 4
Generator and motor bearings	2 to 3
Machine tool bearings	2 to 4

PROPORTIONS OF AUTOMOBILE CRANK SHAFT BEARINGS

A. G. Kessler and G. W. Lewis, in the *American Machinist* of September 29, 1910, give the results of an investigation of the proportions of some 30 representative American automobile engines.

The diameter of the front crank shaft bearing was found to vary from 1 1/2 to 2 1/4 in., and is found from the equation:

$$B_1 = 0.53\,D - 5/8 \text{ in.},$$

where B_1 = diameter of the front bearing in inches,

and D = diameter of the engine cylinder in inches.

The length of the front bearing is expressed by the equations:

$$L_1 = B_1 + 1\ 1/8 \text{ in.}, \text{ and}$$

$$L_1 = 1.67\,B_1 + 0.95 \text{ in.}$$

The reason for two equations lies in the arrangement of the curves drawn through the points plotted from the dimensions of the engines investigated. It shows that there are two types of shaft in use.

The diameter B_2 of the rear crank shaft bearing is found from the equation:

$$B_2 = 0.53\,D - 5/8 \text{ in.}$$

This is identical with the equation for the diameter of the front bearing, as

it is customary to make all bearing diameters the same on automobile engine crank shafts.

The length of the rear bearing is found from the equation:

$L_2 = 5.33\ D_1 - 5.66$ in.

L_2 = Length of rear bearing in inches.

and D = diameter of engine cylinder in inches, as above.

SELECTION OF THE BEARING METAL

The bearing metal must be selected with reference to the metal in the journal with which it is to run, to the load which it is to carry—it must be strong enough to resist deformation—and in many classes of machinery it must be selected for its resistance to wear. See Section III.

Bronze is ordinarily used in the form of a sleeve or bushing either in a single piece or split in half. If a soft metal lining (babbitt) is used, it may either be poured directly into the housing of the bearing or inserted in a removable shell. Such shells are commonly made of cast iron, although brass is used to a limited extent and cast steel still more rarely.

Standard American railway practice consists in alloying the lining and its bronze shell at their joining surfaces. The bronze surface is carefully cleaned, coated with solder, and the lining poured against it. If the work is well done there is no danger of the lining loosening. There is no mechanical anchorage.

Sometimes different metals are used for the lower half of a bearing and its cap. If the cap takes no load it can be lined with a cheaper, softer metal than the load-carrying box.

ANCHORING OF THE METAL LINING

Babbitt linings must be securely anchored to their shells for if they loosen they are apt to crack and be destroyed. If this does not happen, the slight motion between the lining and its shell may act as pump and take the oil out of the bearing. The devices for anchoring are simple. In a common form the shell has a series of ridges with dovetailed edges around which the metal is cast.

Fig. 12 shows an anchoring arrangement using anchors having a circular cross-section from the practice of the Westinghouse Electric and Manufacturing Company. The patterns are made to leave a plain core in the green sand, to which are added baked anchor cores secured with brads as shown. There are two sizes, 1/2 and 3/4 in.

In bearings that for any reason have to be bored before babbitting, the anchor holes are cast extra deep, so that after boring the holes will be standard depth.

The illustration gives the spacing. Along the straight lips of each half, the anchors should be fairly close together, and as near the edge as they can be cast, so that when the casting is finished the anchor holes will break through.

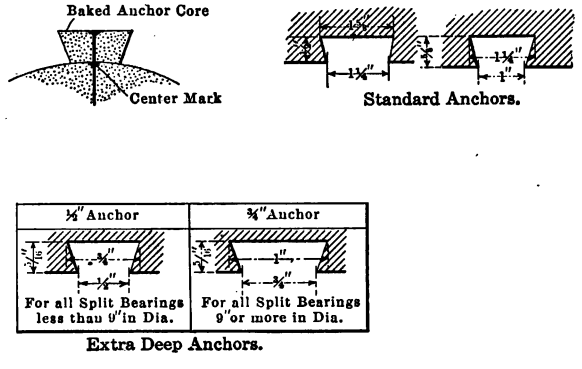


FIG. 12. SIZE AND PITCH OF BABBITT ANCHORS.

ADJUSTING DEVICES TO PRESERVE ALINEMENT

Various devices are used to preserve alinement, or to make bearings self alining within small limits. In some cases, as for instance the spindle bearings of machine tools, the accuracy of the seat controls the alinement. In small electric motors and a large number of shoe machines bushings are forced into place in the frames or housings and are then carefully line reamed. In case a bushing has to be replaced for any cause whatever the process of line reaming must be repeated.

In steam engine work the bearing brasses are usually adjustable by means of wedges or bolts. In electric motors, generators, horizontal turbines, and

some gas engines, the bearings are made with a spherical seat, as shown by Fig. 13.

Bearings for line shafting are usually held on a self-alining ball seat provided with adjusting screws in a vertical direction and with elongated slots in the feet of the hanger for sidewise adjustment. (See Fig. 13.) The self alining feature lies in the ball seat of the box.

Machine countershaft bushings and occasionally more important bearing linings are frequently placed in position on the shaft inside of the housings and

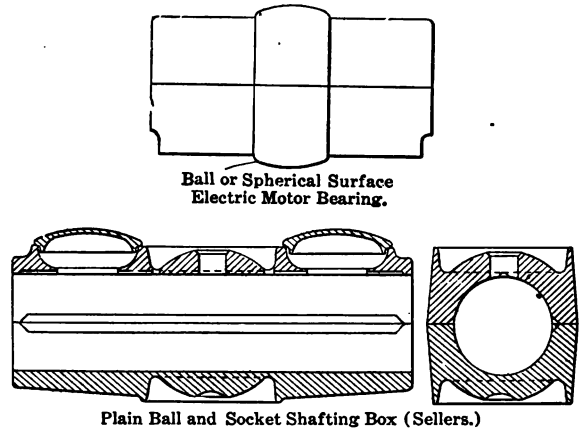


FIG. 13. TYPICAL SPHERICAL ADJUSTING SURFACES FOR BEARINGS.

the space between bushing and housing run with babbitt or an alloy that expands slightly in cooling. This is a simple, cheap, and satisfactory manner to aline and locate such bushings having all of the advantages in manipulation possessed by the babitted box.

On some textile machinery provided with rigid bearings, a slight amount of self-adjustment is obtained by tapering the journal from full diameter at about its centre to a reduction of 0.003 or 0.004 in. at the ends. This double taper allows for a slight amount of swiveling.

CLEARANCE BETWEEN BEARING AND JOURNAL

The space or clearance between the rubbing surfaces is of importance. If too small the coefficient of friction will be very high with a tendency to heat. If too large it will be difficult to maintain a suitable film of oil. The shaft should be freely supported and freely guided in its bearings, without producing any local strain in the material and no local heating through shocks. As a rule the tendency is to make the fit too close. If for any reason a journal

must be tightly fitted in its bearings, which means that the entire circumference of the surface is loaded approximately equally, oil should be supplied by forced lubrication. This condition gives a high amount of friction.

Lasche found that the friction force decreased rapidly with an increasing clearance becoming practically constant when the clearance was 0.06 in. His experiment was with a bearing 10 in. in diameter and 4 in. long. This is an extreme case.

Good operating practice for main engine bearings is a slackness of 0.005 in. for diameters from 6 to 12 in. and for cross-head pins and crank pins up to 6 in. in diameter 0.004 in.

L. D. Burlingame, *Trans. A.S.M.E.*, 1910, gives these limits used by Brown & Sharpe Mfg. Co. for ground cylindrical fits of small size. The tolerances are in decimals of an inch.

RUNNING FITS—ORDINARY SPEED

To 1/2	-in. diameter, inc.	0.00025 to 0.00075	Small
To 1	-in. diameter, inc.	0.00075 to 0.0015	Small
To 2	-in. diameter, inc.	0.0015 to 0.0025	Small
To 3 1/2	-in. diameter inc.	0.0025 to 0.0035	Small
To 6	-in. diameter, inc.	0.0035 to 0.005	Small

RUNNING FITS—HIGH SPEED, HEAVY PRESSURE AND ROCKER SHAFTS

To 1/2	-in. diameter, inc.	0.0005 to 0.001	Small
To 1	-in. diameter, inc.	0.001 to 0.002	Small
To 2	-in. diameter, inc.	0.002 to 0.003	Small
To 3 1/2	in. diameter, inc.	0.003 to 0.0045	Small
To 6	-in. diameter, inc.	0.0045 to 0.0065	Small

SLIDING FITS

To 1/2	-in. diameter, inc.	0.00025 to 0.0005	Small
To 1	-in. diameter, inc.	0.0005 to 0.001	Small
To 2	-in. diameter, inc.	0.001 to 0.002	Small
To 3 1/2	-in. diameter, inc.	0.002 to 0.0035	Small
To 6	-in. diameter, inc.	0.003 to 0.005	Small

GRINDING LIMITS FOR HOLES

To 1	-in. diameter, inc.	Standard to 0.00075	Large
To 2	-in, diameter, inc.	Standard to 0.001	Large
To 3 1/2	-in. diameter, inc.	Standard to 0.0015	Large
To 6	-in. diameter, inc.	Standard to 0.002	Large
To 12	in. diameter, inc.	Standard to 0.0025	Large

Table 34 gives the standard practice of the General Electric Company.

Nominal dimension, inches	Journal		Bearings						Axle linings for ry. motors	
			Horizontal		Vertical		Step			
	Max. diam. in.	Allowable variation below max. diameter	Min. bore in.	Allowable variation above min. bore	Min. bore in.	Allowable variation above min. bore	Min. bore in.	Allowable variation above min. bore	Min. bore in.	Allowable variation above min. bore
3/8	.375	.0005	.377	.001	.376	.001	.3755	.0005	.380	.004
1/2	.500	0005	.502	.001	.501	.001	.5005	.0005	.505	.004
5/8	.625	.0005	.627	.001	.626	.001	.6255	.0005	.630	.004
3/4	.750	.0005	.752	.001	.751	.001	.7505	.0005	.755	.004
7/8	.875	.0005	.877	.001	.876	.001	.8755	.0005	.880	.004
1	1.000	.0005	1.002	.001	1.001	.001	1.0005	.0005	1.005	.004
1 1/8	1.125	.0005	1.128	.001	1.127	.001	1.126	.0005	1.130	.004
1 1/4	1.250	.0005	1.253	.001	1.252	.001	1.251	.0005	1.255	.004
1 1/2	1.500	.0005	1.503	.001	1.502	.001	1.501	.0005	1.505	.004
1 3/4	1.750	.0005	1.753	.001	1.752	.001	1.751	.0005	1.755	.004
2	2.000	.0005	2.003	.001	2.002	.001	2.001	.0005	2.005	.004
2 1/4	2.250	.0005	2.253	.001	2.252	.001	2.251	.0005	2.255	.004
2 1/2	2.500	.0005	2.503	.001	2.503	.001	2.501	.0005	2.505	.004
2 3/4	2.750	.0005	2.754	.002	2.753	.002	2.7515	.0005	2.755	.004
3	3.000	.0005	3.004	.002	3.003	.002	3.0015	.0005	3.005	.004
3 1/2	3.500	.001	3.504	.002	3.504	002	3.5015	.0005	3.507	.004
4	4.000	.001	4.005	.002	4.004	.002	4.002	.001	4.007	.004
4 1/2	4.500	.001	4.505	.002	4.504	.002	4.502	.001	4.509	.004
5	5.000	.001	5.006	.002	5.005	.002	5.0025	.001	5.009	.004
5 1/2	5.500	.001	5.507	.002	5.505	.002	5.503	.001	5.511	.004
6	6.000	.001	6.009	.002	6.005	.002	6.003	.001	6.011	.004
7	7.000	.001	7.011	.002	7.006	.002	7.0035	.001	7.012	.004
8	8.000	.001	8.012	.003	8.006	.003	8.004	.002	8.013	.004
9	9.000	.001	9.013	.004	9.007	.004	9.0045	.002	.....	.....
10	10.000	.0015	10.014	.005	10.007	.005	10.005	.002	.....	.....
11	11.000	.0015	11.015	.005	11.008	.005	11.0055	.002	.....	.....
12	12.000	.0015	12.016	.005	12.008	.005	12.006	.002	.....	.....
13	13.000	.0015	13.016	.005	13.009	.005	13.0065	.002	.....	.....
14	14.000	.0015	14.016	.005	14.009	.005	14.007	.002	.....	.....
15	15.000	.0015	15.016	.005	15.010	.005	15.0075	.002	.....	.....
16	16.000	.0015	16.016	.005	16.010	.005	16.008	.002	.....	.....
17	17.000	.0015	17.018	.005	17.011	.005	17.008	.002	.....	.....
18	18.000	.0015	18.018	.005	18.011	.005	18.008	.002	.....	.....
19	19.000	.0015	19.018	.005	19.012	.005	19.008	.002	.....	.....
20	20.000	.0015	20.018	.005	20.012	.005	20.008	.002	.....	.....
21	21.000	.002	21.018	.005	21.013	.005	21.008	.002	.....	.....
22	22.000	.002	22.020	.008	22.013	.005	22.008	.002	.....	.....
23	23.000	.002	23.020	.008	23.013	.005	23.008	.002	.....	.....
24	24.000	.002	24.020	.008	24.013	.005	24.008	.002	.....	.....
25	25.000	.003	25.020	.008	.....	.....	.....	.....	.....	.....
26	26.000	.003	26.020	.008	.....	.....	.....	.....	.....	.....
27	27.000	.003	27.022	.008	.....	.....	.....	.....	.....	.....
28	28.000	.003	28.022	.008	.....	.....	.....	.....	.....	.....
29	29.000	.003	29.022	.008	.....	.....	.....	.....	.....	.....
30	30.000	.003	30.022	.008	.....	.....	.....	.....	.....	.....
31	31.000	.003	31.022	.008	.....	.....	.....	.....	.....	.....
32	32.000	.003	32.024	.010	.....	.....	.....	.....	.....	.....
33	33.000	.003	33.024	.010	.....	.....	.....	.....	.....	.....
34	34.000	.003	34.024	.010	.....	.....	.....	.....	.....	.....
35	35.000	.003	35.024	.010	.....	.....	.....	.....	.....	.....
36	36.000	.003	36.024	.010	.....	.....	.....	.....	.....	.....

TABLE 34.—ALLOWANCES AND LIMITS FOR JOURNALS AND BEARINGS.

PROVISIONS FOR LUBRICATING

American lubricating oils and greases are by no means standardized as to quality and constituents. It is, therefore, necessary for the designer to consider the possibility of a very poor lubricant being used, and prepare for that condition. American machinery builders do not specify what oil shall be used except by the expression "a good oil for the purpose."

Therefore, at the outset, the designer must prepare for the contingency of a poor oil or grease.

As to the method of applying the lubricant there is a wide choice and it is one of the most important features of design. At the same time it is one that is ordinarily neglected; even a superficial reading of the pages of technical periodicals will show many protests on the part of users of machinery against inadequate oiling devices; even the elementary principles of feeding oil to a bearing are ofttimes ignored. On page 18 it is pointed out that a properly fitted and designed bearing will suck oil against a considerable head, provided it is introduced at a proper point. It has also been pointed out that the oil film in a well lubricated bearing is under considerable pressure. Obviously oil by a gravity feed cannot enter a bearing at this latter point. As a general principle oil fed by gravity must be introduced into a bearing at about the point of least pressure.

The common methods may classified as follows:

Ordinary or imperfect lubrication.	Oil.....	From a squirt can through an open oil hole.
		From a squirt can upon a felt washer covering an oil hole.
		From a squirt can through an oil hole cover to the oil hole beneath.
		From an oil cup through a wick (syphon feed).
		From a sight-feed oil cup.
		From a pad pressed against the journal.
		From an oil well by moving rings or chains or by vanes fastened to the journal.
		From an oil well by centrifugal means.
	Grease...	From a compression grease cup.
		From a stick of grease pressed against the journal.
Perfect lubrication.....	Oil.....	From a pump or gravity system under pressure (15 pounds up) (the oil may or may not be artificially cooled).
		From a bath in which the journal dips.
		From a reservoir in which the bearing is submerged or from which it receives a flood under very little pressure (2 to 5 pounds).

The relative values of several methods of lubrication were found by Goodman, *Proc. Inst. C. E.*, Vol. LXXXIX, page 447, to be:

Bath	Saturated pad	Ordinary pad	Syphon
1.....	1.32	2.21	4.2

The numerical quantities indicate the frictional resistances referred to the best—bath lubrication—as 1.

For all gravity feeds, the point of application should be at about the point of minimum pressure in the oil film.

With forced feed, practice indicates that the point of application should be at an angular distance of 45 degrees from the vertical in a direction counter to the direction of rotation. This is indicated by the diagram Fig. 14.

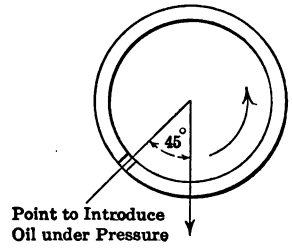


FIG. 14. LOCATION TO INTRODUCE OIL WITH FORCED LUBRICATION.

DISCUSSION OF THE METHODS

The common open oil hole has a serious objection; it is an opening to admit dirt and grit into the bearing as well as oil. With the felt plug it is considerably improved as dirt is excluded and the felt acts as a small reservoir from which oil can drip. (See Fig. 15.) Many self-closing oil hole covers are on the market and serve their purpose until they are broken off or lost. On some kinds of machinery they are used as ornaments.

The oil pad in contact with the journal is a universal method for lubricating railroad car bearings in Europe, and is extensively used for line shafting. It is open to the objection that in time it will become glazed on the surface in contact with the moving journal and lose its capillarity and ability to transfer oil from the well below to the journal above. A remedy is to take it out and soak it in kerosene or gasoline to remove the accumulation of dirt.

The pad itself is often made of a fabric having a face of wool velvet and a back of cotton. This is fastened to a block of wood, and pressed against the journal by springs which lead down to the oil. In some designs, narrow ledges of the wood back fit the journal to reduce the wear on the pad. The arrangements are satisfactory but are open to two objections in America, the expense and difficulty of adapting them to standard equipment.

The standard method of lubricating American car axle journals is by wool waste packing. The wool is elastic even when oil-soaked. It has a disadvantage of less capillary power than cotton and of short length fibers. Cotton, however, loses its elasticity when wet.

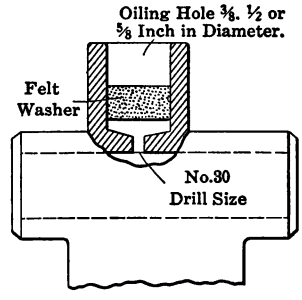


FIG. 15. OPEN OILING HOLE WITH FELT WASHER.

Recent experiments have been made with a woven fabric supported close to the edges of the brass and hanging down in a catenary form into the oil. The pressure against the journal is very slight, and long wicks bring up the oil. The results are promising.

Ring or chain lubrication is the preferred form for better classes of machinery where the heat from the bearing can be dissipated without artificial means as, for instance, in electric generators and motors. Fig. 16 is representative of the arrangement. The rings hang loosely on the journal, rotate with it, pass around the outside of the box, and dip into a reservoir of oil beneath; thus they move in the same direction as the journal and carry oil to its upper surface. A certain amount finds its way to the rubbing surfaces and lubricates the bearing. The rest is merely transferred from one side of the reservoir to the other.

Lasche states that there is a limit of speed beyond which rings cannot be used successfully, commonly said to be the speed at which they begin to "dance" on the shaft instead of rotate. In his experiments he found that, considering the amount of oil delivered to the bearings, this maximum speed was about 2,000 r. p. m., and at about 2,500 revolutions the amount of oil delivered decreased very rapidly with a further increase of speed.

American practice, however, uses rings at journal speeds as high as 3,600 r. p. m. The "dancing" is stopped by increasing the weight of the ring and depth of submersion. After a journal speed of from 600 to 800 r. p. m. has been reached, the rings do not increase in speed for an increase of journal speed. Good proportions are a ring diameter equal to two journal diameters and a submersion in the oil equal to one-half the journal diameter. Allowing for a suitable clearance around the rings and the size and capacity of the oil well are determined.

The curves of Fig. 17 are from Lasche's experiments and may be used as a guide in estimating the amount of oil delivered by rings to the rubbing surfaces of the bearings.

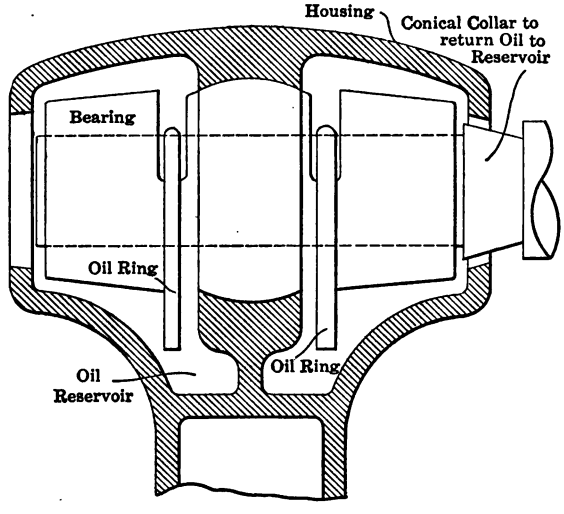


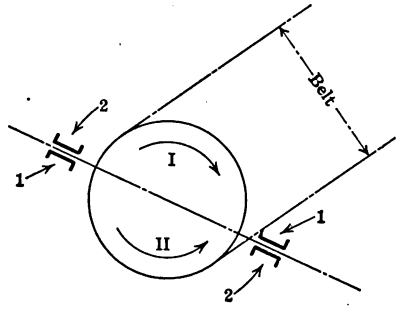
FIG. 16. RING OILING BEARING.

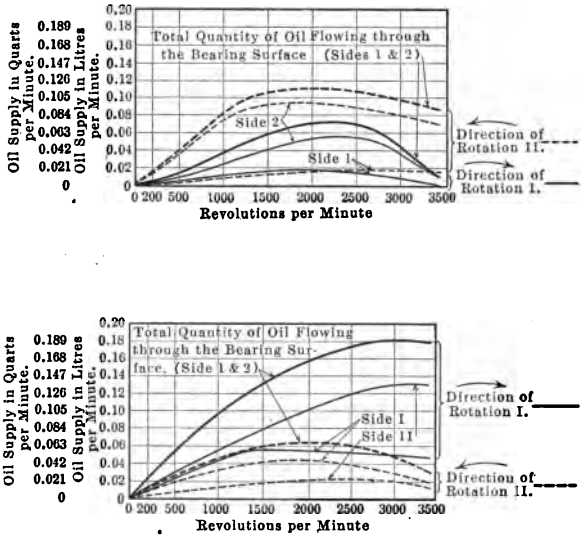
These curves are especially remarkable in that they show how widely different quantities of oil are delivered for similar bearing conditions. When the journal revolved in the direction of the hands of a watch, the quantity of oil flowing was only 1/2 to 2/3 of that which flowed when the journal revolved in the opposite direction. Further, in both directions of rotation the quantity of oil which flowed from the farther side was only about 1/3 of that which escaped from the side nearer the driving belt. When the journal revolved in the more favorable direction and the oil entered at the unloaded side, 0.19 pint per minute flowed through the bearing. When it revolved in the other direction the quantity was only 0.12 pint per minute.

That the relationship of these quantities must not be taken as indicating a

law is shown by the second bearing tested in which the quantities of oil, depending upon the direction of rotation, happened to be in a reverse order.

To work well the rings should be smooth and free from sharp edges and





CONDITIONS: Steel shaft, 3.6 inches in diameter; bearing I of gun metal in one piece 10.4 inches long; bearing II of white metal, split, good mineral oil, two rings immersed 1.6 inches; rings six inches in diameter.

FIG. 17. OIL DELIVERY CAPACITY OF OIL RINGS AT VARIOUS SPEEDS.

corners that may catch and interfere with their free motion. Also the passages in which they lie must be free from projections upon which they might catch. Many split steel rings have been used but are open to the objection of possible

catching. Less commonly they have been die cast in halves from white metal and the parts fastened together by small screws through a halved joint. The preferred ring is continuous, and made either of a hard white metal or brass.

Artificial cooling is seldom resorted to with ring oiling, as there is no good way in which the oil can be cooled. Bearings have been installed with a serpentine coil of pipe in the oil well through which cold water was circulated, but such cases are rare.

There are no data available as to the amount of oil which oil ring reservoirs should contain. The quantity should be great enough so that all particles will have sufficient time to give up their heat to the surrounding metal of the reser-

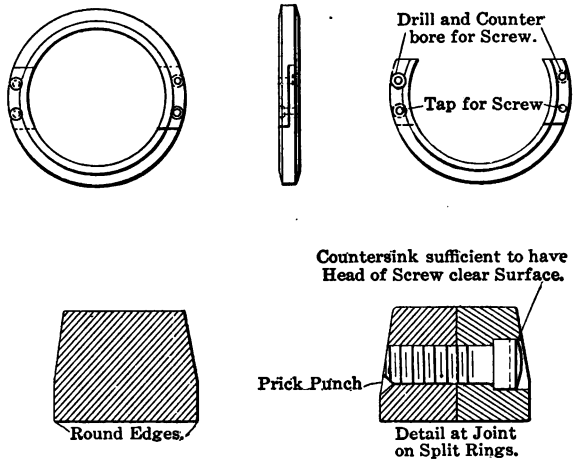


FIG. 18. DESIGN OF SPLIT OIL RING.

voir before they are again brought in contact with the shaft. In some instances, originally hot running bearings have been made safer by merely increasing the quantity of the oil.

Good practice places a limit of 4 in. as the greatest distance that an oil ring can be expected to successfully distribute oil; that is, if a bearing is 8 in. long, one oil ring at the center is sufficient. If the bearing is longer than 8 in. and shorter than 16 in. two rings must be used, and so on in proportion.

Fig. 18 shows an approved form of split ring.

Centrifugal oilers are frequently used with step bearings. They consist of a rotating vane, or series of vanes, that pump the oil from a well into which it drains by gravity, through a system of pipes to the bearings.

There is a growing tendency toward an increased use of greases. These are fed through cups of two general types. In one the cover of the cup is screwed down at intervals by hand upon the contained grease, thus putting it

under pressure and forcing it into the bearings. In the second type the mass of grease is under constant pressure from a spiral spring, or from a weight.

Bath lubrication is not very common. Here the journal dips into a well of

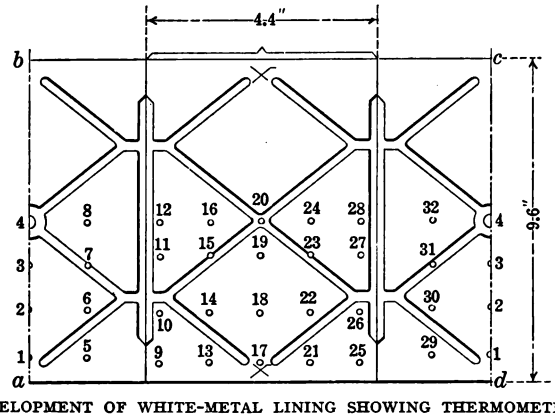


FIG. 19. DEVELOPMENT OF WHITE-METAL LINING SHOWING THERMOMETER LOCATIONS.

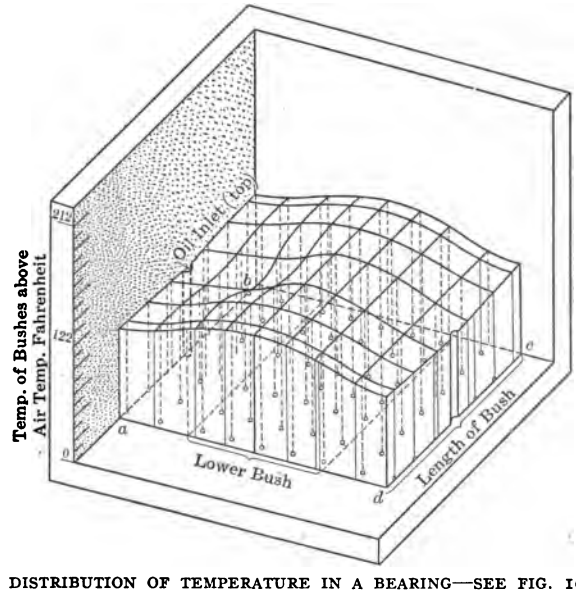


FIG. 20. DISTRIBUTION OF TEMPERATURE IN A BEARING—SEE FIG. 19 (*Lasche*).

oil, and thus has an opportunity to take up all that it will. This is the method of lubrication used by Tower in his celebrated friction experiments.

Flooded lubrication is the supplying of oil to bearings by means of a pump, but at no very great pressure. A typical example is the system used in oiling

the interior bearings of shafts and gears in certain constant speed drive milling machines; also some forms of steam turbine bearings.

Forced lubrication consists in supplying oil to bearings under pressure usually from 15 to 25 lb. per square inch for horizontal bearings. As a rule the oil is cooled by artificial means, although this is not inherent in the system. It is important that the oil shall pass through the bearing at such a rate of flow that it has time to absorb the heat of friction and take it away. This is the reason for introducing cool oil at a point not far distant from that of maximum pressure. In this way intense local heating is avoided. Fig. 19 is reproduced from Lasche's work and shows the development of a white metal lining. The numbered points show where thermometers were brought very close to the inner surface of the lining, and temperature readings taken.

Fig. 20 shows in a solid diagram the observed temperatures. The minimum temperature was at 12, and the maximum at about 19, where the pressure was greatest.

OIL GROOVES

Oil grooves should be designed according to the following principles:

A, The general direction of grooves should be at right angles to the direction of motion. B, They should stop short of the ends of the bearing, to prevent leakage. C, For conditions of considerable speed no groove must pass lengthwise of the bearing in the region of greatest pressure; even a diagonal or circumferential groove should not be used on the pressure side as this will to some extent at least reduce the oil film thickness. It is quite possible that for very slow speed bearings grooves in the pressure area are best. D, The edges of all oil grooves and the meeting edges of split boxes must be carefully rounded over.

Fig. 21 shows the oil grooving practice of the Westinghouse Electric and Manufacturing Company. The grooves vary from 3/32 to 1/4 in. in width and from 3/32 to 1/8 in. in depth.

If grooves are cut clear through to the end, oil leakage will take place. At the same time, if a journal and its bearings are very closely fitted, it may be necessary to scratch or score the surface from the end of the groove out in order to provide an air vent. The grooves at the end of the lower half of the bushings are not brought together, but are separated by some little distance. This is in the region of the greatest oil pressure. If this area is cut across by oil grooves, it interferes with the formation of a proper film. At the meeting point of the halves of the bearing a large V-shaped groove is sometimes cut, which serves as a reservoir of oil, tends to relieve the side pressure on the journal, and aids materially in good lubrication.

At the same time there is a wide diversity of opinion in regard to grooves in bearings. Tradition says put them in, and this is usually done. Yet some engineers claim that there should be no grooves in the half of the bearing that

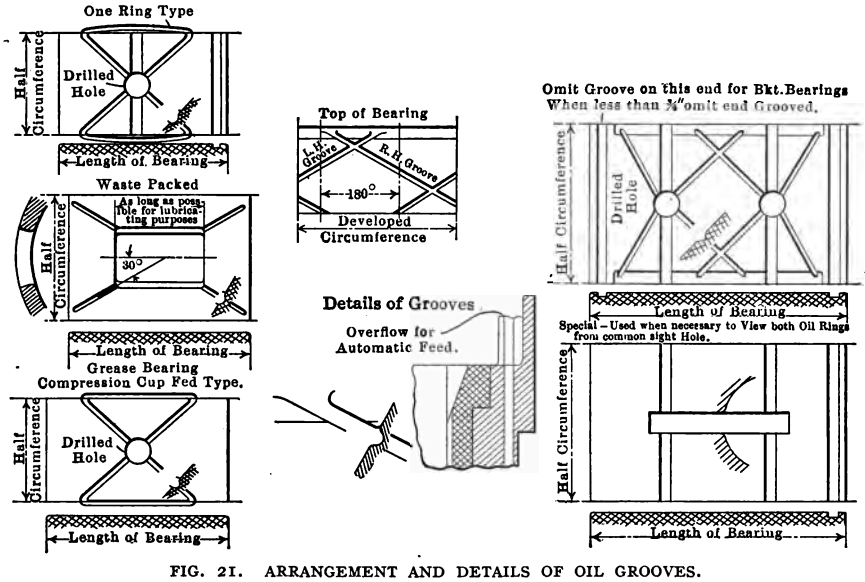


FIG. 21. ARRANGEMENT AND DETAILS OF OIL GROOVES.

supports the load. Milling the box at the split is favored as this tends to form a small reservoir.

Oil grooves are by no means confined to the bearings, but are often cut in the journals, as shown by Fig. 22 from the practice of the Steptoe Shaper Company. Journals thus made are run by them in cast iron bearings with success. It is the

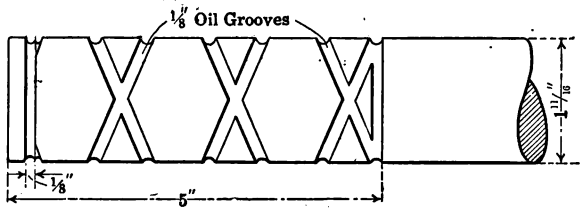


FIG. 22. ARRANGMENT OF OIL GROOVES IN A JONRNAL.

practice of one English builder of heavy engines to mill a flat spot lengthwise of the journals to aid in their lubrication. This undoubtedly distributes the oil over the entire surface of the bearing, and while successful at low speed, is probably questionable for higher speeds, as the coming of the flat spot to the region

of maximum pressure must tend to break down the oil film at each revolution. In machine tools the Osgood system of oil grooving has been used to a considerable extent. It consists in cutting closed spirals, either single or double, in the lining.

PROVISIONS FOR RADIATING AND DISSIPATING HEAT

The heat radiating and dissipating capacity of a bearing must be considered if the engineer or designer is dealing with comparatively high pressures and high velocities. It has been pointed out that the customer usually determines the maximum temperature at which a bearing may run, 140° F. being a common limit. As we have seen, the coefficient of friction for all rough calculations may be taken as constant for the higher commercial speeds. This being so, the maximum limit of speed is determined by the amount of heat produced by friction in a unit of time, which must not be greater than the amount which can be dealt with by radiation and dissipation in the same unit of time. Thus an increase in the speed of the shaft under these conditions is only allowable when the excess of frictional heat resulting therefrom can be carried away by air drafts or oil or water cooling.

Lasche conducted a long series of experiments to determine the heat dissipation from journal bearings. From his results Fig. 23 has been plotted with a transformation of the factors into English units. Three curves are shown. The first is for a bearing surrounded by a thin cast-iron housing so that the bearing surface and the upper surface of the housing can be considered as at the same temperature. In this case it is better to assume that the radiation reaches its lowest limit. Curve 2 is for a standard type of bearing in still air and is the most useful of the three. It can be taken as representing an average for the radiating capacity of ordinary bearings and used as a basis from which to calculate friction work and to determine whether or not a bearing with its natural capacity of radiation can dissipate the heat liberated by friction or whether artificial cooling should be resorted to.

Curve 3 shows radiation from well ventilated bearings with large masses of iron such as are found in turbo-generator construction. The unit work of friction or energy liberated in the form of heat, measured in foot pounds, is given by the formula $f\,w\,V$, in which f is the coefficient of friction, w the unit pressure in pounds per square inch of projected area, and V the rubbing velocity in feet per minute, or letting H equal the rate of radiation per square inch of projected area expressed in foot pounds per minute;

$$H = f\,w\,V.$$

If in the design of a journal bearing the value of the coefficient of friction, f can be determined for the particular working conditions of lubrication, then from the above equation the unit of radiation can be determined by computation.

If we then compare the unit thus found with the radiation as traced from Fig. 23 for the determined permissible rise in temperature of the bearing, it can at once be seen whether the natural radiation is sufficient to maintain a suitable temperature, or some artificial means of cooling must be employed.

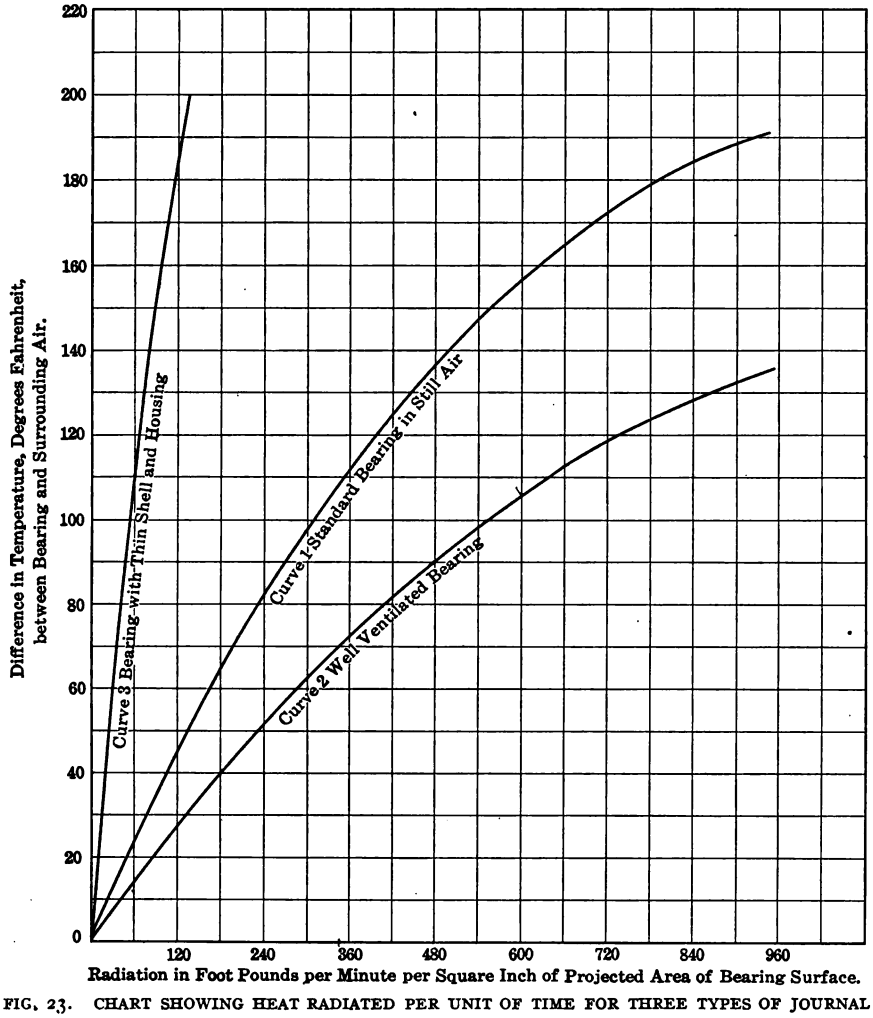


FIG. 23. CHART SHOWING HEAT RADIATED PER UNIT OF TIME FOR THREE TYPES OF JOURNAL BEARINGS.

Good judgment must be exercised in making use of these curves, for the conduction of heat to a bearing along its shaft, as in steam generating machinery, may play an important part in the amount of heat that must be dissi-

pated. Again, if the bearing is surrounded by large masses of iron, with a large amount of exposed radiating surface, the bearing will get rid of more heat than if the surrounding walls are comparatively thin and the radiating surface comparatively small. It is sometimes necessary merely to increase the size of the oil reservoir. Again, an increase in the extent of the surface of a bearing housing may be all that is required to cause a bearing to run within its permissible rise in temperature. Another factor is the proximity of rotating

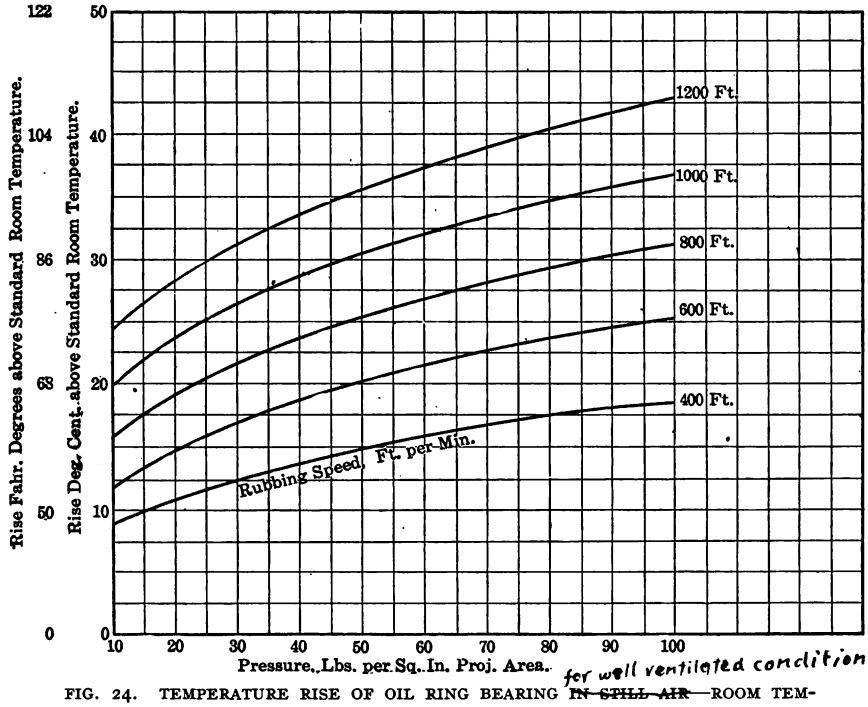


FIG. 24. TEMPERATURE RISE OF OIL RING BEARING ~~IN STILL AIR~~ ROOM TEMPERATURE 25 DEGREES CENTIGRADE, 77 DEGREES FAHRENHEIT.

engine, generator, or motor parts, which set up windage and cause a current of air to pass over the bearing. All of these factors, as well as any others which directly affect the radiation or dissipation of heat must be taken into account in determining the temperature at which a given bearing will run in service. The underlying principle may be stated thus: The work of friction per unit of bearing area must not be greater than the rate of radiation for the same unit area, and for the determined working temperature.

Lasche[1] developed an equation expressing the relationship of the coefficient of friction, unit pressure, and rise in temperature, which transformed into English

[1] For a valuable discussion of this equation see *Elments of Machine Design*, Kimball and Barr, page 247.

units is: $fw\,(t-32)=51.2$, in which f is the coefficient of friction, w the pressure in pounds per square inch, and t the final bearing temperature.

By transformation

$$fw=\frac{51.2}{t-32}$$

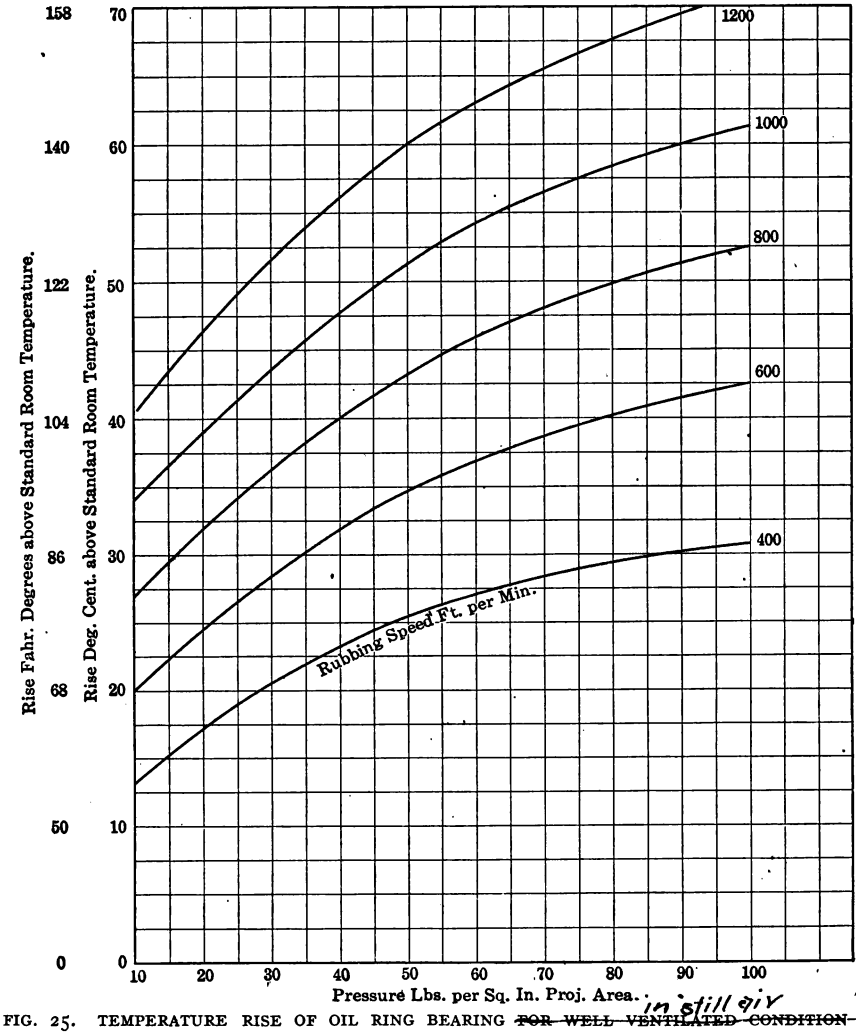


FIG. 25. TEMPERATURE RISE OF OIL RING BEARING ~~FOR WELL VENTILATED CONDITION~~ ROOM TEMPERATURE 25 DEGREES CENTIGRADE, 77 DEGREES FAHRENHEIT.

In the section dealing with values of the coefficient of friction it was shown that for conditions wherein a perfect oil film can be formed the frictional resistance for a given velocity is practically a constant and independent of the pressure. Thus,

as far as ordinary designing is concerned, we may assume that the coefficient of friction for perfect lubrication and all velocities greater than 500 ft. per minute is a constant, and the equation given above may be used without serious error.

By adding the factor V to each side the right hand member becomes equal to the frictional loss per unit of projected area expressed in foot pounds per minute, as below:

$$f w V = \frac{51.2\ V}{t - 32}$$

This equation can now be used to compute the heat liberated by friction in a perfectly lubricated high speed journal bearing for a given velocity in feet per minute at a given working temperature in degrees Fahrenheit.

This gives us a unit of radiation to compare with the curves of Fig. 23 to determine whether or not artificial cooling must be used.

The General Electric Company has carried on an extensive series of tests to determine the rise in temperature for bearings of a type used by them in electrical machinery and under various conditions; the results of these experiments are given in the curves of Figs. 24, 25, and 26. The first is for a ring oiling bearing running in still air, as in a tightly closed room with an air temperature of 77° F. Fig. 25 is for the same kind of bearing well ventilated. Fig. 26 is for a bearing running with forced lubrication and three different rates of oil feed. In each case the lubricating oil was what is known as "dynamo and shafting oil," having a specific heat of 0.4, a weight per gallon of 7.27 lb., and a viscosity at 70° F. of 80.3 Doolittle. Curves 24 and 25 are for a range of speed from 400 to 1200 ft. per minute. The curves of Fig. 26 are for rubbing speeds up to 3,000 ft. per minute and for four different pressures per square inch of projected area, namely, 35, 50, 75, and 100 lb. For pressures and speeds beyond the limit of this chart it is wise to resort to water jacket cooling to remove the liberated heat.

RULES FOR BEARING DESIGNS

This section on bearing design can be closed in no better way than to give the rules for bearing designs as adopted by the Committee of Mechanical Design of the General Electric Company, issued in connection with the preceding charts.

These rules are applicable to the design of the great majority of generator and motor bearings.

The temperature of any bearing having a given radiating surface is determined by the amount of work lost by friction. This work (in foot pounds per minute) $= f\ W\ V$, where,

f = coefficient of friction,

W = load on the bearing in pounds,

V = surface speed in feet per minute.

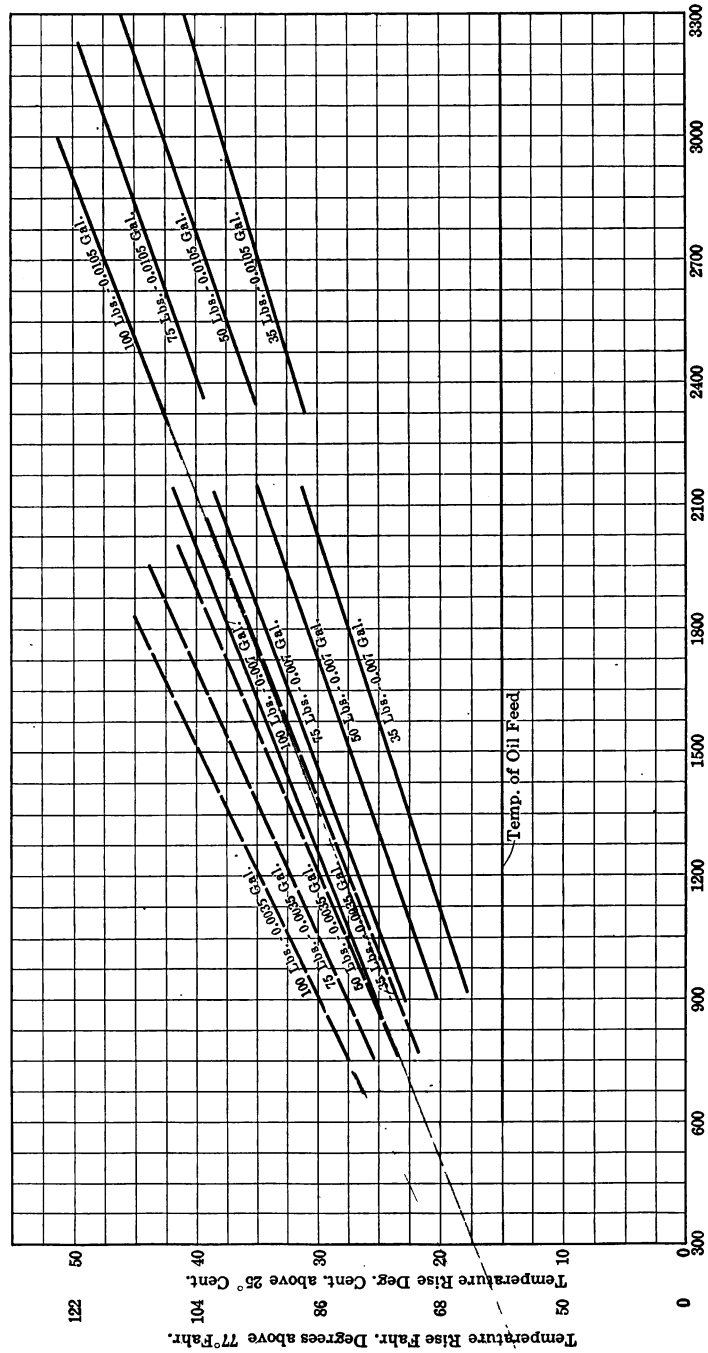


FIG. 26. RELATION BETWEEN RUBBING SPEED AND RISE IN TEMPERATURE FOR FORCED LUBRICATION AND THREE RATES OF OIL FEED—ROOM TEMPERATURE 25 DEGREES CENTIGRADE, 77 DEGREES FAHRENHEIT.

Assuming that the load remains fixed it is evident that the work lost and therefore the amount of heat generated becomes smaller as f and V become smaller; or, in other words the smaller the bearing diameter for a given load the less it will heat. Another advantage of small diameter is that the coefficient of friction decreases with the surface speed. It is, therefore, very desirable in laying out bearings to keep the diameter as small as possible, consistent with sufficient strength of shaft and suitable deflection of the journals both inside and outside the bearings. It is also very desirable to so dimension bearings that they are fairly well loaded, in order to avoid bulky machines and also because the coefficient of friction rises quite rapidly when the load is less than 50 lb. per square inch of projected area.

When calculating the projected area of any bearing, especially if it is to be heavily loaded, the amount of space lost through the drain grooves at both ends must be deducted. This is particularly important when the length of the bearing is small in proportion to the diameter.

It is also necessary—and this applies to all forms of lubrication—that there be no sharp corners on the edges of the oil distributing grooves or channels, but that these be gradually cased off so that the oil can be drawn in between the journal and the bearing. Sharp corners are invariably oil wipers and often absolutely prevent proper lubrication.

The heat generated in any bearing may be dissipated:

1. By radiation from the housings and conduction by the shaft.
2. By forcing cooled oil through the bearing.
3. By surrrounding the bearing by some form of water jacket.

Other forms of artificial cooling are possible but these described under 2 and 3 are the most commonly used.

1.—BEARINGS WITHOUT ARTIFICIAL COOLING

Bearings of this type are usually lubricated by oil rings or similar devices or by gravity feed. It is essential that an abundant supply of oil be delivered to all parts of the bearing by suitably arranging the channels so that a perfect film will be maintained at all times between the journal and bearing, and that there is no opportunity for the oil forming this film to escape through openings or grooves at the points of greatest pressure and thus allow the metals to come in contact.

The heat generated in bearings having no artificial cooling is conducted away and radiated by the housings. The great variation in the design of bearing housings and the different conditions of ventilation, etc., make it extremely difficult to predetermine the ultimate temperature of such bearings with any great accuracy, and it is always necessary to allow a considerable margin of safety.

Fig. 11 covers the range of pressure and speed ordinarily permissible in this type of bearing, while Figs. 24 and 25 show the ultimate temperatures for different speeds and loads. These curves were made up from the readings obtained from special bearings and afterward checked by the test records of a large number of machines—both of the pillow block and shield types—which have gone through the testing department during the last few years. Fig. 25 shows the temperatures to be expected under the most unfavorable conditions, that is, of a bearing so situated that no current of air can circulate about it, and therefore cooled by radiation only. There is, however, a considerable circulation of air about most machines, due to the fanning action of the revolving parts, and the ultimate temperatures to be expected in such cases are shown on the curves of Fig. 24. These curves apply to the great majority of open generators and motors, both of the pillow block and end shield types. When the machine is enclosed, or the free circulation of air in any way interrupted, higher temperatures will result, until finally the conditions of Fig. 25 are reached.

A part of the heat of the bearings of motors and generators is usually conducted away by the shaft and radiated by the spider and other revolving parts. When machines are totally enclosed or are connected to other machinery whose temperature is high, heat may be transmitted through the shaft to the bearing, thus raising the latter's temperature, and due allowance must be made for this.

2.—COOLING BY FORCED LUBRICATION

When bearing pressures and speeds are unusually high it is often necessary to force oil under pressure into the bearings and advantage is often taken of this to keep the heating down by artificially cooling the oil.

The temperature of bearings is also sometimes kept down by forcing cooled oil through them, when otherwise forced lubrication is not necessary. This method, although used to a very considerable extent, is usually not as efficient as a water jacket.

For all practical purposes, it may be considered that the entire heat generated is taken away by the oil, and it is therefore possible to predetermine the bearing temperature with considerable accuracy. Fig. 26 shows the ultimate temperature of bearings using the quantities of oil most commonly pumped through, and with the assistance of these curves the necessary amount can be determined. Intermediate speeds and pressures can be easily interpolated.

For pressures and speeds beyond the limits of this table, it is advisable to resort to water jacket cooling.

In arranging bearings for this form of lubrication, care must be taken to force the oil to the point where the work is being done, as otherwise the oil coming from the bearing may be comparatively cool, while the bearing itself

is much too warm. In addition to this, it means that an excessive amount of oil must be pumped through the bearing requiring unnecessarily large pumps, pipings, etc. Experiments with such bearings show that when oil begins to run out of the ends quite freely, nothing is gained by forcing through a larger quantity.

3.—COOLING BY WATER JACKET

A properly designed water jacket will carry away a very much larger amount of heat than will any form of forced oil lubrication, as the specific heat of water is very much higher. As is the case of forced oil lubrication the ultimate temperature of a well designed water jacket bearing can be very accurately determined. It is essential that the pipes or channels be located as close as possible to the surface of the lining where the work is being done, in order that the heat generated may be absorbed without danger of damage to the lining. Water circulated at some distance away from the lining surface is of comparatively little assistance, as heat may be generated so rapidly that the lining will be destroyed before the heat strikes the jacket. The water passages must also be so arranged that an even and continuous circulation is kept up in all parts.

With properly constructed passages, it is safe to assume that heat may be removed at the rate of from 3 to 5 horse power for each gallon of water circulated per minute, and where the conditions are unusually good and the jacketing carefully arranged from 10 to 12 horsepower per gallon can be dissipated.

With water jackets any suitable method of lubrication may be used which will insure at all times a good film of oil between journal and bearing.

SECTION VI

Lubricants for Bearings

The subject of lubricants for bearings is one of great complexity. The chemical analysis of oils and greases is very difficult and is work that can be successfully carried on only by a specially trained chemist. Many oils are mixed or blended and for the user there are no simple tests by means of which he can determine just what the constituents of a particular kind of oil are. At the best he can only make a few simple physical tests and then study the behavior of the oil in use.

The following paragraphs will not touch upon the chemical side, but merely point out some of the important kinds of oils and greases, their applications, and some of the physical characteristics that they should have.

CLASSIFICATION OF LUBRICANTS

All lubricants fall into three classes: oils, greases, and solids.

Lubricating oils.....	High pressure cylinder oil, Low pressure cylinder oil, Engine oil, Dynamo and shafting oil, Machinery oil for heavy duty, Machinery oil for light duty, Loom oil, Spindle oil.
Lubricating greases... ...	Non-fluid oil, Soft grease, Hard grease.
Solid lubricants.....	Graphite, Sulphur, Talc.

The oils are again divided into three divisions as regards origin, mineral, vegetable and animal. The mineral oils are petroleum products and in this country are spoken of as of two kinds, oils with a paraffine base and oils with an asphalt base, the Pennsylvania product being representative of the first class and the Texas product of the second. Oils from certain localities have come to be recognized as standards for some purposes, as, for instance, the best cylinder oils are supposed to be made from Franklin stock.

Among vegetable oils, colza, rape seed, olive, and palm oils are good lubricants. Castor oil has good lubricating qualities but oxidizes and gums. Among animal oils, lard, tallow, sperm, and whale are all used both pure and for compounding with mineral stock.

Graphite is the only solid lubricant that is used to any extent, and it is recognized as one of our most valuable lubricants. It is employed in a solid form fitted into recesses in bearing linings, in powdered form mixed with oil and in the form of deflocculated graphite mixed with oil or water. This last form is a special product of the Acheson Graphite Company. Sulphur was formerly employed for emergency use on over-heated bearings, prior to the common use of graphite. Ground talc or soapstone free from grit has had a limited use for the same purposes.

Water has been used as a lubricant, particularly for the steps of vertical water turbines and vertical steam turbines. Its action, however, is not that of a true lubricant. In the vertical steam turbine and some water turbines it serves as a means to separate the plates of the steps. It is pumped in under pressure and floats the load. In other water turbines having wood block and metal cup bearings it removes the heat liberated by friction and prevents the bearing from burning and cutting.

Many other kinds of oils are used but they are often looked upon as adulterants; for instance, cotton-seed oil is frequently mixed with lard oil. As mentioned above, their detection is very difficult and should only be undertaken by the trained chemist. Just what oils may be used as adulterants at a given time depends largely upon the relative market values of the oils.

Turning to physical tests, the oil user is most largely interested in the qualities of viscosity and body. The viscosity of an oil is the property that determines the rate of free flow.

The property, body, is difficult to define. The user gets his idea of it by dipping his thumb and forefinger into the oil and rubbing them together. The degree of oiliness or greasiness is an indication to him of its body. It has been defined as that property of an oil that influences the change in viscosity when the oil is under pressure; or as that property that influences the intensifying of the viscosity in that portion of the oil film within the region of attraction of the surface molecules of the metals of bearing and journal.

Professor Kingsbury, *Trans. A. S. M. E.*, 1903, page 147, gives these relations of viscosity and body.

With increase of	Where the viscosity is effective, the coefficient of friction	Where the body is effective, the coefficient of friction
Pressure	Decreases	Increases.
Speed	Increases	Decreases.
Temperature	Decreases	Increases.
Viscosity	Increases	Decreases.
Body	.....	Decreases.

From this it is seen that the effects of body and viscosity are in nearly all respects diametrically opposite and it must necessarily be very difficult to derive reliable information regarding the lubricating values of oil from friction tests in which the effects of viscosity and body are not separately recognizable. Thus we have but very little information that is of value on the body of oils. As a general rule, mineral oils have much less body than animal and vegetable oils. Thus of a mineral oil and lard oil of approximately the same viscosity the lard oil would have decidedly the greater body. On the other hand, some cylinder oils of mineral origin greatly exceeding lard oil in viscosity will also have a greater body.

It is not uncommon to mix a small percentage of animal oil with mineral oil in order to increase the body. For illustration, turn to the latter part of this section where specifications for oils used by the United States War and Navy Departments are given.

During the past few years there has been a tendency to use more and more grease for the lubrication of machinery bearings. In comparing the relative qualities of a grease and an oil, the former must be tested at the running temperature of the bearing where it is to be used. In general the lower the speed the thicker should be the lubricant. With heavy pressures and low speed it is difficult to form a film of oil and maintain it. This is specially true when the machine is started after being shut down. While idle the oil is squeezed out from between the journal and bearing surfaces and there is metallic contact which produces abrasion as soon as motion begins with an attending rise in temperature and thinning of the oil if that is used. For such work greases are especially suitable, as the solid or semisolid substance is not squeezed out from the journal in the way a liquid is acted upon.

Professor Goodman, *Proc. Inst. C. E.*, Vol. LXXXIX, page 432, gives comparative results of three oils and one grease. The nominal pressure per square inch was 85 lb.

Lubricants	Natural coefficient of friction	Coefficient of friction after standing 16 hours
Machinery oil	0.0084	0.192
Valve oil A	0.0252	0.147
Thick and viscous B	0.0329	0.1715
Hard solid grease	0.0350	0.090

It will be noted that the coefficient of friction after standing is only about half as great for the grease as for the oils.

On the other hand, a grease should not be used unless the conditions for feeding it to the bearing are such that it will reach all parts of the rubbing surfaces. An incident given by W. H. Booth, *American Machinist*, Vol. XXXII part 2, page 790, bears on this point.

"A set of pumps driven by a worm and wheel were required to be tested for general sufficiency. The worm was below the wheel and there was a thrust bearing in a narrow cylindrical recess out of the oil casing which inclosed the worm and wheel. A thick oil was filled into the casing and the test was made. The thrust bearing soon grew unbearably hot, and it looked as if the pump would be condemned. Quite suddenly the box went comparatively cool and no further heating took place. The cooling coincided with the thinning or liquefying of the oil. At first this was so very thick that it could not freely enter the thrust chamber against the tendency of the worm to draw the oil away from the thrust recess and pile it up on the opposite side of the casing, but this piling up ceased when the oil became hot and thin, for it could then get well into the thrust collars, and these cooled at once, and the test was declared good. A thinner oil would have run from the start in all likelihood, and this is confirmed by the experience of many, that a thin oil is a better lubricant than a thick oil up to the point where the pressure begins to squeeze the oil from between the surfaces."

From Tower's experiments, *Proc. Inst. M. E.*, 1883, the following comparison of five oils and one grease is taken:

Lubricant	Mean resistance, pounds	Per cent.
Sperm oil	0.484	100
Rape oil	0.512	106
Mineral oil	0.623	129
Lard oil	0.652	135
Olive oil	0.654	135
Mineral grease	1.048	217

The second column gives the actual frictional resistance at the surface of the journal per square inch of bearing area at a speed of 300 revolutions per minute, and for nominal loads from 100 to 310 lb. per square inch. The percentage figures may be taken to represent the relative thickness or body of the various oils, and in their order, if not in numerical proportion, the relative weight carrying ability. Thus it is seen that the sperm oil has by far the greatest lubricating value and the least weight carrying ability.

It must be recognized that these values, however, hold only for the conditions

of investigation. For greater pressures or higher temperatures the relative order of excellence might easily be changed with some of the heavier oils leading.

Another very important quality of any lubricating oil is that it shall be free from acids and alkalies. Animal and vegetable oils must be watched more closely in this particular than mineral oils. As a rule mineral oils have no action on metals unless they contain sulphur, and then there is no liability of harm except at high temperatures. Among the organic oils lard oil and pure animal oils in general have little action; on the other hand, olive and vegetable oils generally produce corrosion.

Other points to be investigated in lubricating oils are the flash and fire points, the amount of contained water, the amount of tarry and suspended matter, the volatility and action under saponification. For good practice in regard to these points see the United States War Department specifications in the latter part of this section.

QUANTITIES OF OIL USED

As is to be expected, there are few data available for the quantity of oil used for bearing lubrication. The variation in type of bearing, rubbing speed, pressures, quality of the oil and its method of application, all effect this question.

As representative of American practice in the lubrication of steam engine cylinders, Table 35 is reproduced from *Power*, Feb. 15, 1910, page 303. A glance at the last column shows that the average is about one pint for every million square feet of cylinder surface rubbed over.

For bearings lubricated by forced pressure and water cooled 1/50 gal. per square inch of projected area per minute is good practice.

For bearings with flooded lubrication and cooling of the oil 1/10 gal. per square inch of projected area per minute is frequent practice.

For the step bearings and guide bearings of Curtis vertical steam turbines Table 36 can be considered as presenting average practice.

No.	Estimated h. p.	Description of engine	Cylinders	Stroke	Revolutions per minute	Steam pressure	Sq. ft. rubbed over per hour
1	1,350	Corliss triple expansion	20–34–52	60	65	.....	1,084,000
2	675	Cross compound	h. p. 26	48	81	.....	265,800
3	675	Cross compound	l. p. 52	48	81	.....	531,600
4	975	Cross comp. St. Louis Corliss	22–44	48	85	140	706,500
5	975	Corliss compound	24–44	60	65	.....	694,500
6	650	Hamilton Corliss cross comp	20–36	44	100	140	644,400
7	650	Hamilton Corliss cross comp	20–36	44	100	140 100° sup	644,400
8	575	Corliss compound	18–34	48	85	.....	555,900
9	875	Corliss tandem compound	24–42	48	65	110	539,600,
10	450	Russell cross compound	16–30	24	150	.....	433,300
11	650	Cross comp. condensing	18–36	42	72	.....	428,100
12	290	Cross compound	h. p. 18	36	86	120	146,000
13	290	Cross compound	l. p. 34	36	86	120	276,000
14	200	Harris standard cross comp	h. p. 16	18	200	110	150,800
15	200	Harris standard cross comp	l. p. 28	18	200	110	263,900
16	400	Cross comp. St. Louis Corliss	14–28	42	85	.....	393,000
17	450	Corliss cross comp	18–30	36	84	.....	380,000
18	290	Phoenix Iron Works	14–24	18	210	130	376,100
19	200	Russel tandem comp	13–20½	20	210	.....	368,200
20	225	Corliss cross compound	h. p. 16	36	84	.....	126,800
21	225	Corliss cross compound	l. p. 30	36	84	.....	237,500
22	225	Buckeye tandem comp	12–21	21	200	.....	363,100
23	450	Ingersoll cross compound	18–30	24	100	.....	302,000
24	200	American ball compound	10½–20½	12	285	.....	277,600
25	180	Westinghouse automatic comp.	11–19	11	300	.....	259,300
26	160	Cross comp. Meyer valves	11–18	14	165	115	175,400
27	160	Tandem comp. piston valve	10½–18	18	115	115	172,100
28	70	Ingersoll cross comp	8–12	12	150	.....	94,300
29	450	Piston valve	30	48	95	150	357,800
30	450	Double eccentric corliss	30	48	84	150	318,000
31	290	Corliss	24	48	102	.....	308,100
32	290	Harris Corliss	24	48	100	.....	302,000
33	290	.....	24	42	93	120	245,500
34	200	Greene	20	44	100	.....	230,200
35	160	Sioux Corliss	18	42	105	.....	207,900
36	290	Harris Corliss	24	48	65	.....	196,200
37	200	Corliss	20	36	102	150	196,200
38	130	New Brown	16	36	125	110	188,800
39	250	Atlas single valve automatic	22½	30	105	110	183,500
40	160	Ames	18	16	230	.....	173,700
41	160	Corliss	18	36	100	.....	169,800
42	130	High speed automatic	16	16	250	115	167,800
43	130	High speed automatic	16	16	250	115	167,800
44	150	Piston valve engine	17½	16	220	80	161,500
45	130	Slide valve throttling governor	16	30	126	95	158,200
46	120	Single valve auto. Fitchburg	15½	14	275	.....	156,200
47	160	Atlas four valve	18	15	220	125	155,900
48	130	Harris Corliss	16	36	100	65	150,900
49	130	.....	16	38	92	.....	146,600
50	100	.....	14	14	280	.....	143,200

TABLE 35.—STEAM ENGINE CYLINDER LUBRICATION (*Power*)

No.	Oil used — Name or description	Price per gal.	Amount used per hr., pints	Cost per. hr., cents	Pints per estimated 1000 h. p. hours	Cents per estimated 1000 h. p. hours	Pints per million sq. ft. rubbed over
1	.....	.28	0.69	2.42	0.511	1.79	0.636
2	No. 725 cyl. comp.....	.75	0.466	4.37	0.69	6.46	1.76
3	Capitol oil.....	.35	0.666	2.91	0.99	4.33	1.25
4	No. 650 dark valve cyl.....	.50	0.104	0.65	0.107	0.67	0.147
5	.....	.28	0.621	2.17	0.639	2.23	0.89
6	"600 W.".....	.60	0.21	1.57	0.323	2.42	0.33
7	Oil of beeswax; 600 fire test cyl.stock and acidless tallow	.90	0.255	2.87	0.392	.....	0.40
8	.....	.28	0.465	1.63	0.81	2.84	0.84
9	Heavy body comp.....	.40	0.2	1.00	0.229	1.14	0.37
10	"600 W.".....	.60	0.25	1.87	0.555	4.16	0.58
11	Improved high pressure cyl...	.75	0.111	.....	0.171	.....	0.26
12	Best grade cyl.....	.60	0.645	4.34	2.22	15.80	4.42
13	Best grade cyl.....	.60	0.323	2.42	1.11	8.32	1.17
14	Harris h. p. valve.....	.60	0.184	1.38	0.92	6.90	1.22
15	Rarus.....	.....	0.131	.....	0.65	.....	0.50
16	No. 650 dark valve.....	.50	0.09	0.56	0.225	1.41	0.23
17	"600 W.".....	.60	0.417	3.13	0.927	6.95	1.10
18	.....	.43	0.167	0.90	0.58	3.10	0.44
19	"600 W.".....	.60	0.167	1.25	0.835	6.25	0.45
20	Best grade cyl.....	.60	0.645	4.34	2.86	21.45	5.10
21	Best grade cyl.....	.60	0.323	2.42	1.43	10.72	1.36
22	"600 W.".....	.60	0.25	1.87	1.111	8.33	0.69
23	"600 W.".....	.60	0.167	1.25	0.371	2.78	0.55
24	.....	.....	0.273	.....	1.365	.....	0.98
25	No. 725 cyl. comp.....	.75	0.083	0.78	0.461	4.32	0.32
26	Best high grade mineral.....	.50	0.012	0.075	0.075	0.47	0.07
27	High grade mineral.....	.50	0.037	0.23	0.231	1.44	0.21
28	"600 W.".....	.60	0.083	0.62	1.185	8.89	0.88
29	"Cyl. oil No. 10.".....	.25	0.417	1.30	0.927	2.90	1.16
30	"Cyl. oil No. 10.".....	.25	0.5	1.56	1.111	3.47	1.57
31	.....	.....	0.091	.....	0.314	.....	0.29
32	.....	.....	0.091	.....	0.314	.....	0.30
33	.....	.60	0.369	2.77	1.272	9.55	1.50
34	.....	.....	0.91	.....	0.455	.....	0.39
35	"600 W.".....	.60	0.125	0.94	0.781	5.86	0.60
36	.....	.....	0.19	.....	0.655	.....	0.97
37	"Cyl. oil No. 10.".....	.25	0.167	0.52	0.835	2.61	0.87
38	Harris high-press valve.....	.60	0.118	0.88	0.908	6.81	0.63
39	Buckeye cyl.....	.28	0.15	0.525	0.600	2.1	0.82
40	Capital oil.....	.35	0.22	0.96	1.375	6.00	1.27
41	.....	.90	0.1	1.12	0.625	7.04	0.59
42	High grade mineral.....	.50	0.015	0.094	0.115	0.72	0.089
43	High grade mineral.....	.50	0.22	0.14	0.169	1.06	0.13
44	Light colored oil.....	.....	0.03	.....	0.2	.....	0.19
45	Capitol.....	.35	0.1	0.44	0.769	3.36	0.63
46	.....	.28	0.138	0.48	1.15	4.02	0.88
47	Rarus cyl.....	.....	0.094	.....	0.587	.....	0.60
48	Harris std. grade.....	.60	0.15	1.12	1.153	8.64	0.99
49	Compounded oil.....	.43	0.15	0.81	1.153	6.20	1.02
50	Model cyl.....	.....	0.045	.....	0.45	.....	0.31

TABLE 35.—STEAM ENGINE LUBRICATION (*Power*).

No.	Estimated h. p.	Description of engine	Cylinders	Stroke	Revolutions per minute	Steam pressure	Sq. ft. rubbed over per hour
51	240	.....	22	23	106	.....	140,500
52	160	Wright Corliss.....	18	42	70	85	138,800
53	130	Nordberg Corliss.....	16	36	90	.....	135,800
54	100	Robt. Armstrong automatic....	14	14	260	125	133,500
55	130	Corliss.....	16	42	75	.....	132,100
56	100	Corliss.....	14	36	100	.....	132,000
57	200	Slide valve.....	20	30	80	80	125,800
58	160	Greene.....	18	30	85	.....	120,200
59	100	Variable speed, St. Louis Corliss.	14	36	90	135–140	118,800
60	85	Atlas automatic.....	13	18	190	125	116,800
61	110	Ames.....	15	14	212	.....	116,600
62	100	Automatic piston valve.....	14	14	223	.....	114,500
63	130	Nordberg Corliss.....	16	32	85	110	113,800
64	70	Ideal.....	12	12	300	100	113,100
65	70	Buckeye	12	21	165	.....	108,900
66	100	Corliss.....	14	36	80	.....	105,800
67	70	Atlas single valve automatic....	12	18	180	110 very wet	101,800
68	70	Corliss.....	12	36	90	.....	101,800
69	70	Bates Corliss.....	12	36	88	100	99,500
70	70	St. Louis Corliss.....	12	36	85	135–140	96,350
71	70	New Brown.....	12	34	90	80	96,200
72	50	.....	10	14	260	80	95,400
73	100	Atlas.....	14	20	130	80	95,400
74	50	McEwen.....	10	12	300	.....	94,400
75	70	Fitchburg.....	12	30	100	90	94,400
76	70	Slide valve.....	12	14	210	150	92,500
77	40	Ball.....	9	12	300	.....	84,750
78	50	Atlas.....	10	12	240	.....	75,400
79	50	Center crank.....	10	15	170	.....	66,800
80	85	.....	13	17	105	100	60,900
81	50	Slide-valve.....	10	12	250	100	31,410

TABLE 35.—*Continued.*—STEAM ENGINE CYLINDER LUBRICATION (*Power*)

No.	Oil used — Name or description	Price per gal.	Amount used per hr., pints	Cost per hr., cents	Pints per estimated 1000 h. p. hours	Cents per estimated 1000 h. p. hours	Pints per million sq. ft. rubbed over
51	"Eureka cyl."	.....	0.105	.....	0.437	.....	0.75
52	Capitol	.35	0.125	0.55	0.781	3.41	0.90
53	Valvaline	.75	0.2	1.87	1.538	14.38	1.47
54	"600 W."	.60	0.15	1.12	1.5	11.22	1.12
55	.....	.28	0.138	0.48	1.061	3.71	1.04
56	"600 W."	.60	0.06	0.45	0.6	4.50	0.45
57	.....	.....	0.018	.....	0.09	.....	0.14
58	"600 W."	.60	0.03	0.22	0.187	1.46	0.25
59	No. 650 dark valve cyl.	.50	0.045	0.28	0.45	2.82	0.38
60	"600 W." vacuum cyl.	.60	0.067	0.50	0.788	5.90	0.57
61	Capitol	.35	0.11	0.48	1.0	4.37	0.94
62	High grade heavy	.50	0.125	0.78	1.25	7.8	1.09
63	Dark heavy cyl. with graph.	.....	0.115	.....	0.885	.....	1.01
64	Capitol cyl.	.35	0.083	0.36	1.186	5.18	0.73
65	.....	.....	0.1	.....	1.428	.....	0.92
66	.....	.....	0.049	.....	0.49	.....	0.47
67	Buckeye cyl.	.28	0.6	2.1	8.57	30.0	5.94
68	.....	.....	0.0835	.....	1.19	.....	0.82
69	Capitol cyl.	.35	0.2	0.87	2.857	12.48	2.01
70	No. 650 dark valve cyl.	.50	0.045	0.28	0.643	4.01	0.47
71	Franklin oil	.75	0.1	0.94	1.428	13.38	1.04
72	Capitol cyl.	.35	0.055	0.24	1.1	4.81	0.58
73	Capitol cyl.	.35	0.2	0.87	2.0	8.74	2.10
74	W. P. Miller's cyl. comp.	.75	0.04	0.37	0.8	7.50	0.42
75	.....	.45	0.055	0.31	0.786	4.42	0.58
76	Flake graph. with eng. oil	.....	0.083	.....	1.185	.....	0.90
77	Pomo oil	.....	0.167	.....	4.175	.....	1.97
78	.....	.60	0.045	0.34	0.9	6.75	0.60
79	.....	.....	0.083	.....	1.66	.....	1.24
80	"600 W."	.60	0.125	0.94	1.47	11.02	2.05
81	Premium valve oil	.....	0.091	.....	1.82	.....	2.90
				Averages	1.014	6.19	0.97

TABLE 35.—*Continued.*—STEAM ENGINE CYLINDER LUBRICATION (*Power*).

Rating		Gallons oil per minute					Pressure per square inch				
Kilo-watts	R. p. m.	Step	Guide	Gear	Total	Step	Baffler drop	Pump-less line	Gear and guides	Unbalanced rev. wt.	Step block O. D., I. D.
750	1800	3	1.5	2.5	7	290	60	350	120	10800	8x5 3/4
1500	1500	4	1.75	2.25	8	320	65	385	120	19700	11x8
3750	900	9	2.0	5.0	16	525	100	625	120	65000	16x9 1/4
5000	750	12	3.0	6.0	21	825	120	945	120	101000	16x10
9000	750	15	4.0	8.0	27	900	150	1050	120	148000	18½ x 10 3/4
14000	750	18	4.0	10.0	32	900	150	1050	120	190000	20x13 1/2
15000	750	20	4.0	12.0	36	850	150	1000	120	216000	21x15
20000	750	25	4.0	14.0	43	950	150	1100	120	233000	21x15

TABLE 36.—VERTICAL STEAM TURBINE LUBRICATION.

FILTRATION OF LUBRICATING OILS

Oil after having passed through machinery bearings seems to lose a large proportion of its lubricating power, besides being contaminated with foreign particles, such as metallic dust, carbon, dirt and the like. It is a difficult matter to restore it to its original purity and value. The usual shop method is by filtration, but the filtered oil is seldom used alone as a lubricant. It is usually mixed with a certain amount that is fresh.

The three common methods of filtration are by capillary action, by gravity filtering, or by pressure filtering. Of these the second is the most common.

G. W. Bissell describes a capillary filter in the *Trans. A.S. M. E.*, Vol. XVII, page 293. "The oil filter that I used consists of two shallow rectangular tin trays and some wide lamp-wicks. One of the trays, slightly smaller than the other, is supported within it on two blocks, which raise it an inch or so from the bottom of the larger pan. The wicks are laid in the upper pan, so as to hang over its edges into the lower pan. The oil to be filtered is poured into the upper pan. A drip cock in the lower pan, and a cover for the whole, complete the apparatus. No quantitative tests have been made, but the results, as far as the eye can judge, are good."

Gravity filtration consists in passing the used oil through mediums, such as mineral or animal wool, bone dust, vegetable or animal charcoal, hair felt, absorbent cotton and the like. Heat is usually applied, and the final condition of the filtered oil depends upon the cleaness of the material and the rate of its flow. It is doubtful if any of these materials can completely remove finely divided metallic particles.

Pressure filtering is only used for very large quantities of oil.

SIMPLE TESTS FOR GREASES.

Professor G. W. Lewis writing in the *American Machinist*, gives a few simple tests of greases as follows:

There are two general types of greases; the so-called mineral, and the animal or tallow greases. Mineral greases should be made from a high grade refined petroleum oil and so treated that the least possible amount of foreign substances be used to bring it to a solid state. Lime soap is generally used and the mixture emulsified by mechanical sitrring or blowing air through the mixture. Some manufacturers use resin, resinous oils, graphite, soap-stone wax, talc, powdered mica, and asbestos, which are introduced into the grease for the purpose of creating an artificial body.

Tallow greases should consist of some hard animal or vegetable fat, such as tallow, with a small amount of mineral oil. Here the mixture is solidified by the use of a soap, in this case a potash soap being usually used. The melting temperature of tallow grease is from 50 to 75 degrees lower than the melting temperature of the mineral grease. Some very inferior greases are made from oils which are the waste of soap factories and sewage disposal systems.

All greases look good when the can is opened and nearly all will work well on a bearing for a short time, and then a poor grease will cause the bearing to heat, which decomposes the grease into oil and soap, especially if the per cent. of soap is high, and impurities have been used. Any adulterants used continually accumulate in the bearings as the grease melts away, and cause gumming and sometimes scoring of the bearing.

The consumer, to protect himself, should make the following simple tests on any grease he contemplates using, test for acids, alkalies, volatile matter, filling, water and gumming.

The test for free acid is best made by melting a small quantity of grease and applying blue litmus paper. The test for alkali is made in the same way by applying red litmus paper.

The test for volatile matter, such as naphtha or benzine, is made by heating a measured quantity to 150° to 200° F. and determining by weight the amount volatilized.

The test for filling, or the amount of soap or other material, used to form a body, can be partially determined by melting a test-tube of the grease to a liquid state and noting the amount of residue or cloudiness in the bottom of the tube, which gives an idea of the amount of soap or other material used in bringing the grease to a solid state. A more accurate test is to take a small sample of grease and dissolve in gasoline, then add phosphoric acid in alcohol solution and if any soap is present, it will precipitate and give a relative idea of the amount in the grease.

To determine the amount of water in the grease, and to see if it will "wash

out," place a small amount in a linen cloth or a common handkerchief, fold in the form of a bag, and place in water and knead for five minutes. In this way the amount of water present can be determined, and also if the grease will wash out, which is very important, especially in paper making and laundry machines where water comes in contact with the bearings.

The simplest method to determine the amount of gumming is to take a sherry glass, put in a few pieces of copper wire, and add concentrated nitric acid, then add the grease in finely divided particles to the solution, and stir until all the grease has been broken up. In the case of a tallow grease, the tallow will form a solid crsut on top of the liquid in about an hour; with a vegetable oil, the oil remaining will be about the consistency of butter, while with a mineral oil, the oil remains in a liquid state. The blackish substance that appears below the oil represents the impurities that cause gumming of the bearing. Next separate the acid from the oil and the black substance and add gasoline to the latter two. This washes out the oil and separates the gumming substances so that a fair idea of the quantity in the grease can be obtained.

PENNSYLVANIA RAILROAD COMPANY SPECIFICATIONS FOR LARD OIL

Two grades of lard oil, known in market as "Extra" and "Extra No. 1," will be used, the former principally for burning and the latter as a lubricant.

The material desired under this specification is oil from the lard of corn fed hogs, unmixed with other oils, and containing the least possible amount of free acid. Also from October 1 to May 1 it should show a cold test not higher than 40° F. Oil from lard of "mast" or distillery fed hogs does not give good results in service, and should never be sent. Also care should be taken to have the oil made from fresh lard. Old lard gives an oil that does not burn well, and also gums badly as a lubricant. The use of the so-called neatsfoot stock, either alone or as an admixture in making the "Extra No. 1" grade, is not recommended. Neatsfoot oil is used by the Railroad Company when the price will admit, but it is preferred to have it unmixed.

Shipments must be made as soon as possible after the order is placed. Also shipments received at any shop after October 1 will be subjected to cold test and rejected if they fail, unless it can be shown that the shipment has been more than a week in transit.

Shipments of the "Extra" grade will not be accepted which

I. Contain admixture of any other oils.
II. Contain more free acid than is neutralized by 4 c. c. of alkali as described in the printed method.
III. Show a cold test above 45° F. from October 1 to May 1.
IV. Show coloration when tested with nitrate of silver as described below.

Shipments of "Extra No. 1" grade will not be accepted, which,

I. Contain admixtures of any other oils.

II. Contain more free acid than is neutralized by 20 c. c. of alkali as described in the printed method.

III. Show a cold test above 45° F. from October 1, to May, 1.

The cold test and the amount of free acid must be determined in accordance with P. R. R. standard methods, printed copies of which will be furnished upon application.

The nitrate of silver test is as follows: Have ready a solution of nitrate of silver in alcohol and ether, made on the following formula:

Nitrate of silver	1 gram.
Alcohol	200 grams.
Ether	40 grams.

After the ingredients are mixed and dissolved, allow the solution to stand in the sun or in diffused light until it has become perfectly clear; it is then ready for use and should be kept in a dimly lighted place and tightly corked.

Into a 50 c. c. test-tube, put 10 c. c. of the oil to be tested (which should have been previously filtered through washed filter paper), and 5 c. c. of the above solution, shake thoroughly and heat in a vessel of boiling water 15 min. with occasional shaking. Satisfactory oil shows no change of color under this test.

UNITED STATES NAVY SPECIFICATIONS

Following are the specifications for lubricating oil for marine machinery as issued by the United States Navy Department, September 13, 1906, and still in force.

1. Must be a properly compounded oil to form a homogeneous compound that will not separate under varying temperatures, and must consist of a pure mineral oil, and not more than 30 per cent. nor less than 20 per cent. of suitable non-drying fixed oils as may be best suited for lubrication.

2. The compounded oil must be free from rosin, tar, drying oils, sulphur, asphaltic or tarry bodies, soaps or oil thickners, water, grit, dirt or other suspended matter; and must be free from mineral acid, and must not contain more than 2 per cent. of free oleic acid. The specific gravity to be between 0.915 and 0.927 at 60° F.

3. *As a lubricant* the oil when tested on an oil-testing machine, owned and operated by the Government, having a standard brass bearing of about 9 sq. in. projected area on a polished steel mandrel making about 160 revolutions per minute with a surface speed of about 250 ft. per minute must, to be satisfactory, perform as follows:

(*a*) The temperature of bearing must at no time during the test be permitted to exceed 130° F. Only sufficient oil under test to be applied to prevent excessive friction and heating of bearing.

(*b*) The average load on the bearing for two hours must be at least 300 lb. per square inch of projected area of bearing.

(*c*) The quotient found by dividing the product of the average total pressure on journal and surface speed of journal in feet per minute by the weight of oil in grains used for lubrication in the test must not fall below 325,000.

(*d*) The duration of test must not be less than two hours nor more than two and one-half hours, and but one test will be permitted for any one lot of oil.

4. To be purchased and inspected by weight, the number of pounds per gallon to be determined by the specific gravity of the oil at 60° F. multiplied by 8.33 lb., the weight of a gallon (23 cu. in.) of distilled water at the same temperature.

5. Flashing point must not be below 400° F.

6. Freedom from gumming. Using a half-pint brass oil cup maintained at about 140° F., practically equal quantities of oil must feed through the wick in equal intervals of time for three intervals of eight hours each; the wick to be of zephyr wool, four strands, doubled once. The oil in the cup will be brought to the original level at the beginning of each hour, and not less than 14 oz. avoirdupois must feed through the wick during the first period of eight hours. At the end of test the wick must be clean and the sides of oil cup bright and clean.

7. *Cold Test.*—The oil must flow at a temperature of 32° F.

8. *Freedom from Acid.*—A small quantity when applied to a polished copper plate must not turn the surface of the metal green if allowed to stand exposed to the air for twenty-four hours.

9. Viscosity by Engler viscosimeter. At 90° F. must not be below 685; at 150° F. must not be below 155; at 225° F. must not be below 75, compared with distilled water (49) at 90° F.

10. Oil accepted and paid for under these specifications will be the subject of further tests in actual use on board ship, and any brand of oil failing to afford thoroughly efficient service in these conclusive tests shall not again be considered for naval use.

UNITED STATES WAR DEPARTMENT SPECIFICATIONS

No. 1

ENGINE OIL

This oil is suitable for general lubrication of marine engines. All work except cylinder.

SPECIFICATIONS

Must be a compounded oil composed of 25 per cent. pure acidless tallow oil and 75 per cent. pure mineral oil. Must be free from acid, alkali, and suspended matter, and must satisfactorily pass the following tests:

TESTS

Specific Gravity.—Must not be below .920 nor above .923 at 60° F.

Flash.—Must not flash below 415° F.

Fire.—Must not burn below 470° F.

Viscosity.—Must not be below 8.11 at 50° C. (Engler.)

Cold Test.—Must flow at a temperature of 35° F.

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Oil must not froth nor bump in flash cup when heated.

Saponification —When oil is treated with alcoholic caustic potash must show presence of 25 per cent. tallow oil.

No. 2

CYLINDER OIL (DARK)

This oil is suitable where steam pressures are high and lubricating conditions severe.

SPECIFICATIONS

Must be a compounded oil composed of 5 per cent. pure acidless tallow oil and 95 per cent. pure mineral oil. Must be free from acid, alkali, tarry or suspended matter, and must satisfactorily pass the following tests.

TESTS

Specific Gravity.—Must not be less than .900 nor more than .905 at 60° F

Flash.—Must not flash below 540° F.

Fire.—Must not burn below 600° F.

Viscosity.—Must not be below 3.83 at 100° C. (Engler.)

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Tarry and Suspended Matter.—Put 5 c.c. oil in 100 c.c. stoppered measuring cylinder and 95 c.c. of benzine. Shake well and allow to stand at least 10 minutes, and must be no precipitation.

Volatility.—Heat small quantity of oil on watch glass for two hours at 400° F. There must not be a loss of more than 5 per cent. by weight.

Saponification.—When oil is treated with alcoholic caustic potash must show presence of 5 per cent. tallow oil.

No. 3

CYLINDER OIL (FILTERED)

This oil is suitable where a light colored valve oil is desired.

SPECIFICATIONS

Must be a compounded oil composed of 15 per cent. pure acidless tallow oil, and 85 per cent. pure filtered mineral oil. Must be free from acid, alkali tarry or suspended matter, and must satisfactorily pass the following tests:

TESTS

Specific Gravity.—Must not be less than .891 nor more than .895 at 60° F.

Flash.—Must not flash below 470° F.

Fire.—Must not burn below 540° F.

Viscosity.—Must not be less than 2.30 at 100° C. (Engler.)

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Tarry and Suspended Matter.—Put 5 c.c. oil in 100 c.c. stoppered measuring cylinder and 95 c.c. of benzine. Shake well and allow to stand at least 10 minutes. There must be no precipitation.

Volatility.—Heat small quantity of oil on watch glass for 2 hours at 400° F. There must not be a loss of more than 5 per cent. by weight.

Saponification.—When oil is treated with alcoholic caustic potash, must show a presence of 15 per cent. tallow oil.

No. 4

GAS ENGINE OIL

This oil is suitable for lubrication of gas engines that are water cooled, also for bearings.

SPECIFICATIONS

Must be a pure filtered mineral oil. Must be free from acid, alkali, suspended matter, and satisfactorily pass the following tests.

TESTS

Specific Gravity.—Must not be less than .896 nor more than .899 at 60° F.

Flash.—Must not flash below 400° F.

Fire.—Must not burn below 460° F.

Viscosity.—Must not be less than 3.09 at 50° C. (Engler.)

Cold Test.—Must flow at a temperature of 29° F.

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—Must be unaffected by an alcoholic solution of caustic potash.

No. 5

GAS ENGINE OIL

This oil is suitable for lubrication of gas engine cylinders that are water cooled, also bearings, where an oil of lighter body than No. 4 is desired.

SPECIFICATIONS

Must be a pure filtered mineral oil. Must be free from acid, alkali, and suspended matter, and must satisfactorily pass the following tests.

TESTS

Specific Gravity.—Must not be less than .874 nor more than .877 at 60° F.

Flash.—Must not flash below 385° F.

Fire.—Must not burn below 430° F.

Viscosity.—Must not be less than 4.61 at 50° C. (Engler.)

Cold Test.—Oil must flow at temperature of 33° F.

Acid.—Must not give a reaction of acid on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—Must be unaffected by an alcoholic solution of caustic potash.

No. 6

ENGINE OIL

This oil is suitable for lubrication of high speed engines, dynamos, and high speed work generally.

SPECIFICATIONS

Must be a pure filtered mineral oil. Must be free from acid, alkali, and suspended matter, and pass satisfactorily the following tests.

TESTS

Specific Gravity.—Must not be less than .874 nor more than .878 at 60° F.

Flash.—Must not flash below 375° F.

Fire.—Must not burn below 420° F.

Viscosity.—Must not be less than 2.40 at 50° C. (Engler).

Cold Test.—Must flow at a temperature of 14° F.

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—Must be unaffected by an alcoholic solution of caustic potash.

No. 7

ENGINE OIL

This oil is suitable for very troublesome bearings.

SPECIFICATIONS

Must be a pure filtered mineral oil. Must be free from acid, alkali, and suspended matter, and pass satisfactorily the following tests.

TESTS

Specific Gravity.—Must not be less than .877 nor more than .880 at 60° F.

Flash.—Must not flash below 340° F.

Fire.—Must not burn below 390° F.

Viscosity.—Must not be less than 4.50 at 50° C. (Engler).

Cold Test.—Must flow at a temperature of 32° F.

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—Must be unaffected by an alcoholic solution of caustic potash.

No. 8

ENGINE OIL

This oil is suitable for engine bearings and difficult work where pressure is not extraordinarily high.

SPECIFICATIONS

Must be a pure filtered mineral oil. Must be free from acid, alkali, suspended matter, and pass satisfactorily the following tests.

TESTS

Specific Gravity.—Must not be less than .875 nor more than .879 at 60° F.

Flash.—Must not flash below 380° F.

Fire.—Must not burn below 420° F.

Viscosity.—Must not be less than 2.98 at 50° C. (Engler).

Cold Test.—Must flow at a temperature of 22° F.

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—Must be unaffected by an alcoholic solution of caustic potash.

No. 9

MACHINE OIL

This oil is suitable for shafting and ordinary lubricating duties on light running machinery.

SPECIFICATIONS

Must be a pure filtered mineral oil. Must be free from acid, alkali, and suspended matter, and pass satisfactorily the following tests.

TESTS

Specific Gravity.—Must not be less than .868 nor more than .872 at 60° F.

Flash.—Must not flash below 370° F.

Fire.—Must not burn below 420° F.

Viscosity.—Must not be less than 2.38 at 50° C. (Engler).

Cold Test.—Must flow at a temperature of 14° F.

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—Must be unaffected by an alcoholic solution of caustic potash.

No. 10

MACHINE OIL

This oil is suitable for ordinary machinery where heavy pressure and slow speed call for an oil of heavier body than No. 9.

SPECIFICATIONS

Must be a pure filtered mineral oil. Must be free from acid, alkali, and suspended matter, and pass satisfactorily the following tests.

TESTS

Specific Gravity.—Must not be less than .890 nor more than .893 at 60° F.

Flash.—Must not flash below 390° F.

Fire.—Must not burn below 440° F.

Viscosity.—Must not be less than 4.29 at 50° C. (Engler).

Cold Test.—Must flow at a temperature of 32° F.

Acid —Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—Must be unaffected by an alcoholic solution of caustic potash.

No. 11

ENGINE OIL

This oil is suitable for use in cylinders of kerosene engines.

SPECIFICATIONS

Must be a compounded oil composed of 20 per cent. pure acidless tallow oil, and 80 per cent. pure filtered mineral oil. Must be free from acid, alkali, and suspended matter, and pass satisfactorily the following tests.

TESTS

Specific Gravity.—Must not be less than .884 nor more than .888 at 60° F.

Flash.—Must not flash below 395° F.

Fire.—Must not burn below 430° F.

Viscosity.—Must not be less than 4.06 at 50° C. (Engler).

Cold Test.—Must flow at a temperature of 30° F.

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—When oil is treated with alcoholic caustic potash must show presence of 20 per cent. tallow oil.

No. 12

CYLINDER OIL

This oil is suitable for use in ammonia cylinders of ice and refrigerating machinery.

SPECIFICATIONS

Must be a pure filtered mineral oil. Must be free from acid, alkali, and suspended matter, and satisfactorily pass the following tests.

TESTS

Specific Gravity.—Must not be less than .870 nor more than .875 at 60° F.

Flash.—Must not flash below 370° F.

Fire.—Must not burn below 420° F.

Viscosity.—Must not be less than 2.38 at 50° C. (Engler).

Cold Test.—Must flow at temperature of 10° F.

Acid.—Must not give an acid reaction on polished copper in 24 hours.

Alkali.—Ash must not show an alkaline reaction.

Water.—Must not froth nor bump when heated in flash cup.

Saponification.—Must be unaffected by an alcoholic solution of caustic potash.

No. 13

CYLINDER OIL

This oil is suitable for use in cylinders of Westinghouse Engines.

SPECIFICATIONS

Must be a pure mineral oil, and pass satisfactorily the following tests.

TESTS

Specific Gravity.—Must not be less than .892 nor more than .896 at 60° F.

Flash.—Must not flash below 450° F.

Fire.—Must not burn below 540° F.

Viscosity.—Must not be less than 3.11 at 100° C. (Engler).

Acid —Must not give an acid reaction on polished copper in 24 hours.

Alkali —Ash must not show an alkaline reaction.

Water.—Must not froth not bump when heated in flash cup.

Sponification.—Must be unaffected by an alcoholic sloution of caustic potash.

Tarry and Suspended Matter.—Put 5 c.c. oil in 100 c.c. stoppered measuring cylinder and 95 c.c. of benzine. Shake well and allow to stand at teast 10 minutes; must give only slight precipitation.

Volatility.—Heat small quantity on watch glass for two hours at 400° F. Mtus not loose more than 5 per cent. by weight.

SECTION VII

Design of Sliding Surfaces and Special Bearings

We are indebted to Professor Sweet for a most valuable discussion of the design of flat sliding surfaces in a little book, *Things That Are Usually Wrong*. The following is adapted from that source. The underlying principal is to make the wearing surfaces of equal length. With this proportion experience has shown that even under varying loads and speeds the wear is very small and practically uniform over the entire extent of the contact surfaces. The reason is stated thus: "Things that do not tend to wear out of true do not wear much."

Professor Sweet has used this principle in the cross-head shoes and guides of steam engines, for the wearing surfaces between column and knee and horizontal slide and knee of a milling machine, for the ram and guides of a punch press and for the cross slide of a lathe. In each case the design was successful.

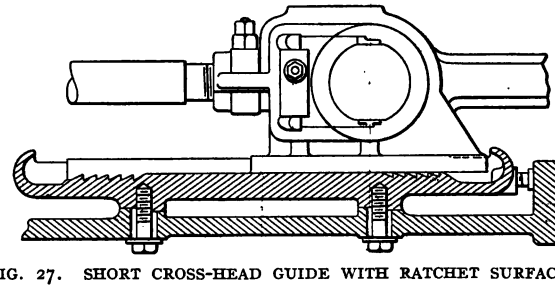


FIG. 27. SHORT CROSS-HEAD GUIDE WITH RATCHET SURFACES.

Fig. 27 shows a cross head and guide with equal length bearing surfaces made by reducing the length of the guide through cutting a ratchet surface at each end. This surface holds the oil pushed along by the end of the head and acts as a reservoir from which lubricant is taken on the return stroke. This construction is said to prevent the splashing and throwing of oil to a much greater extent than a full length guide.

In some classes of machinery, particularly machine tools where the sliding members are short stroked much of the time, excessive local wear may take place either on the sliding member or on the guides. The shaper is an illustration. Here the contact surfaces of ram and guide are sure to wear most at the front end. As a partial remedy, Professor Sweet suggests to cut away the wearing surfaces at the rear increasing the proportion of gaps to lands as the

rear end is approached. Fig. 28 shows how this is done and gives a graphical method of determining the widths of gaps and lands. The shaded portion represents the parts of the surface that have been cut away and the white portion those remaining to take the wear.

The method of laying out is to draw a diagonal AB across the surface to be relieved; then lay off the line BC at a convenient angle with the length of the guide, say 45 degrees. From C measure CD equal to the width of the guide

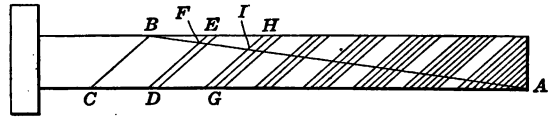


FIG. 28. METHOD OF RELIEVING FLAT WEARING SURFACES.

and draw DE parallel to BC and intersecting AB at F. The vertical distance from F to the upper edge of the slide is the width to be relieved in the next horizontal section of the guide having a length equal to the guide width. This process of laying out is repeated to the end, the widths of the gaps increasing.

In a similar manner the surface of the sliding member should be relieved but sloped in the opposite direction. If the sliding member is very short in comparison to the length of the guide no relief will be necessary.

With regard to the ways of lathes, Professor Sweet says: "There has always been diverse practice and, from time to time, much discussion about the guiding

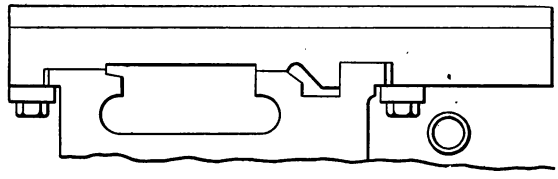


FIG. 29. A DESIGN FOR LATHE WAYS.

of the slide-rests of lathes. The V has the merit of remaining free from lost motion, however much worn, but nothing is more ridiculous than two V's, for the one at the back does no good and costs money. The common flat way is bad because the guiding surfaces are too far apart. The plan adopted by John Lang & Sons, Fig. 29, is much better, the guiding, though by flat surfaces, is at the front where it ought to be, and when all the metal and work is concentrated on the one guide it can be twice as long and four times as efficient."

LUBRICATION OF SLIDING SURFACES

The lubrication of sliding surfaces is difficult owing to the wiping action of the moving member. It is common to oil groove the surfaces and feed oil by

some of the ordinary methods. Unless oil is introduced under pressure the grooves should run across the face at right angles to the direction of motion and should be stopped at some distance from the edges. If thus made they will aid in forming and maintaining a lubricating film. If forced lubrication is used the grooves can be cut along the diagonals of the surface and the lubricant introduced through a hole at their intersection. All of the edges of the oil grooves and the edges of both slides and guides should be chamfered or rounded over, to do away with scraping action.

Planer ways are commonly oiled by a small roller dipping in an oil pocket, the number used depending upon the length of the bed. Professor Sweet says that it would be a great improvement "to use deep pockets and wheels 6 inches or a foot in diameter, and then to pipe the oil from the end pockets to the bottom of the roller pockets and thus make the oiling continuous."

INFLUENCE OF REDUCING THE ARC OF CONTACT

The arc of contact between journal and brass in railroad car axle bearings is made considerably less than 180 degrees. This, of course, reduces the projected area and is sometimes referred to as a peculiarity in design. John Goodman, *Proc. Inst. C. E.*, Vol. LXXXIX, gives the results of experiments and an accompanying empirical mathematical investigation which shows that the force of friction for a given set of conditions can be reduced by decreasing the arc of journal contact within certain limits.

The experiments were made with an alloy bearing composed of 1 part tin and 7 parts copper; it was 4 in. long, 2 in. in diameter, while its width or the length of the chord of contact was varied from 2 in. to 1/2 in. The journal with which it ran was made of manganese steel. Several forms of lubrication were tried. The results show that the oil film is probably at its best for an arc of contact subtended by a center angle of from 80 to 110 degrees.

Table 37 gives the mean results of these experiments.

The empirical formula which agrees very closely with the experimental results is:

$$Log.\ R = \log.\ R_1 + 0.00671\ C_0$$

in which R = the frictional resistance to be determined when the width of the chord is equal to C.

R_1 = the frictional resistance found by experiment, when the width of the chord is C_1.

r = the frictional resistance when the width of the chord = 1 (or 0.01 of the diameter). This is termed the base of the curve.

$C_0 = C - C_1$.

To show the use of this equation, two examples are given as follows: Given a shaft 5 in. in diameter fitted with a semi-circular brass, the total frictional resistance is found by experiment to be 8 lb., it is required to find how much the friction will be reduced by cutting down the chord of the arc of contact to 3 in., the total load is assumed to be unaltered.

Here $R_1 = 8$ lb., log. $8 = 0.90309$.
$C = 3$ in., $= 60$ units.
$C_1 = 5$ in., $= 100$ units.
$C_0 = 2$ in., $= 40$ units.

Then *log.* $R = 0.90309 - 0.00671 \times 40$;
$= 0.63469$
$R = 4.31$ lb.

Thus, by reducing the width of the brass from 5 in. to 3 in., the frictional resistance has been reduced from 8 lb. to 4.31 lb., or 46 per cent.

Given an axle of 5 in. in diameter carrying a load of 7,500 lb. on a semi-circular bearing 10 in. long; the frictional resistance is 50 lb. Assuming that the intensity of the load must not exceed 300 lb. per square inch of projected area, how much can the frictional resistance be reduced by shortening the arc of contact? The equation for the shortest permissible chord is:

Log. R minimum $=$ *log.* $R_1 + 0.00671\ (x - C_1)$ in which the notation is the same as given above, except that x is the length of the shortest permissible chord.

Then $x = \dfrac{7.500}{300 \times 10} = 2.5$ in. $= 50$ units

and *log.* R minimum $=$ log. $50 + 0.00671\ (50 - 100) = 1.3635$
R minimum $= 23.1$ lb.

Thus the frictional resistance on this brass may be reduced from 50 to 23.1 lb. without exceeding the safe intensity of load.

Total load lb.	Length of Chord of Contact									
	2 in.		1.75 in.		1.5 in.		1.0 in.		0.5 inch	
	Angle Subtended									
	180 degrees		122 degrees		97 degrees		60 degrees		29 degrees	
P	f	fP	f	fP	f	fP	f	fP	f	fP
50	0.0441	2.20	0.0458	2.29	0.0418	2.09	0.0210	1.05	0.0126	0.63
100	0.0288	2.88	0.0238	2.38	0.0209	2.09	0.0105	1.05	0.0084	0.84
150	0.0192	2.88	0.0160	2.40	0.0140	2.10	0.0070	1.05	0.0056	0.84
200	0.0159	3.18	0.0121	2.42	0.0106	2.12	0.0055	1.10	0.0042	0.84
250	0.0128	3.20	0.0097	2.42	0.0085	2.12	0.0044	1.10	0.0034	0.84
300	0.0103	3.09	0.0091	2.43	0.0071	2.13	0.0037	1.11	0.0028	0.84
350	0.0085	2.97	0.0070	2.45	0.0061	2.13	0.0033	1.15	0.0024	0.84
400	0.0071	2.84	0.0062	2.48	0.0054	2.16	0.0029	1.16	0.0021	0.84
450	0.0062	2.79	0.0055	2.47	0.0049	2.20	0.0028	1.26	0.0019	0.84
500	0.0058	2.90	0.0049	2.45	0.0044	2.20	0.0029	1.45	0.0017	0.84
550	0.0051	2.85	0.0045	2.47	0.0040	2.20	0.0027	1.48	0.0015	0.84
Mean	.....	2.89	.....	2.42	.....	2.14	.....	1.18	.....	0.82

CONDITIONS: Bath lubricator at 104° F., 233 r. p. m.

TABLE 37.—MEAN RESULTS OF EXPERIMENTS WITH DIFFERING ARCS OF CONTACT.

KNIFE-EDGE BEARINGS

Knife-edge bearings are used in weighing machines, testing machines and scales. J. W. Bramwell, *Engineering News*, Vol. LV, page 653, discusses their design. From his article the following is condensed.

Allowable loads on knife edges vary with the manner in which the pivots or knife edges are held in the levers and the pivot supports of scales, and secured to the bases of weighing machines.

Very little care is exercised in most heavy weighing scales as to the proportion of length of the pivot or the method employed in securing it to its lever. Frequently slender pivots are overhung from the side of the lever, and because of the deflection, even under a moderate load, the extension of the pivot beyond the solid support is practically worthless. The case of the pivot is much worse than the shaft because the local distortion is greater; hence the pivot would touch its support only over a fraction of its length.

In testing machinery greater care is generally given to these details, although serious faults are still prevalent with some machines considered as standard.

The testing machine is nothing but a weighing scale having a straining mechanism, which pulls upon a specimen; the resistance of this specimen is weighed precisely as upon a platform scale.

A defective but cheap practice in fitting pivots to levers of testing machines, particularly those designed to carry a heavy load, is to cast the pivots in the body of the lever using square tool-steel pivots for cores. These pivots have a taper which allows them to be driven out when the casting is cool. In spite of the care used in placing these cores in the mold their correct position will be changed after the casting is poured. To bring them into proper alinement again, considerable skill and care is necessary.

The greater fault of such practice, however, is in the shape of the hole in the casting. The hole being square and its diagonal necessarily vertical, a very bad breaking point is made by the sharp corner. The tendency to break is also increased by the chilled iron immediately in contact with the pivot. The initial strain due to driving the taper pivot in still further reduces the strength of the lever at this point. The hole is frequently enlarged by the wedging action of the pivot under load, and on account of the irregular surface at the upper sider of the pivot it will assume another position and change what should be a fixed ratio—the lever lengths. Allowable loads are variable because of these irregularities.

Loads of 10,000 lb. per inch of length on knife edges are permissible, but the pivot must be flat at its upper portion, normal to the load and supported its whole length with a minimum deflection of parts to ensure reasonable accuracy.

The quality of steel used in both pivots and seats has an important bearing upon the allowable load. In all cases it is essential that a high-grade uniform tool steel be used, having a carbon content of 0.90 to 1 per cent. Such steel will take a very high temper and yet have sufficient ductility to resist sharp blows without crumbling. The temper of the seats should be drawn to a very light straw color; that of the pivots should be *slightly* darker.

An angle of 90 degrees for the knife edge has given good results for heavy loads. For ordinary weighing machinery and most testing machinery 5,000 lb. per inch of length will be found an ample allowance. In a recently built testing machine of 800,000 lb. capacity the two large levers had a load of 10,000 pounds per inch of length. For greater loads the sharp edge is rubbed with an oilstone so that a smoothness is just visible. A pronounced radius of knife edge will decrease the sensibility of the apparatus. The seat, if an angular one, is shaped with a small radius at the intersection of the sides of the angle.

SPINDLE BEARINGS FOR TRAVERSE SPINDLE GRINDERS

The bearings for the spindles of traverse spindle grinders, essentially watch factory tools, are an exception in that under proper conditions of fitting they are run without lubrication. Commercially they are usually run with a very

little watch oil, benzine or kerosene. They show a practical application of the phenomenon of air lubrication.[1]

In diameter, they are about 1/2 in. and run at speeds from 12,000 to 13,000 r. p. m. Both bearings and mating spindles are of hardened steel most carefully lapped as to straightness and fit.

W. H. Sawtell, in the *American Machinist* for August 5, 1909, page 248, writes of the conditions under which these bearings run:

"The traverse spindle when properly fitted will run without a particle of oil and it is common practice to use it in this way. However, it must be kept clean. Use clean tissue paper and alcohol to clean the spindle and bushings, being careful to keep oily fingers from coming in contact with them. If the spindle shows a tendency to cloud up or stick, rub it with a little wad of paper, keeping the spindle revolving and traversing to reach as much of the surface as possible.

"Spindles which will run absolutely dry are rare, almost as rare as the men who understand how to make and use them. Most spindles made commercially are fitted a little free and require the least bit of lubricating. The best method is to wad up tightly a piece of clean waste about the size of a walnut and put on it a couple of drops of watch oil. Apply this to the revolving spindle at the same time traversing back and forth several times. If the spindle appears to gum or cloud, keep at it, and after a few cleanings and oilings it will settle down to business and give no further trouble.

"The fit of these spindles in their bearings is very close, more like the fit of a plug gage and its mating ring than of sliding fits as we ordinarily know them. Under the best of conditions they will run with air lubrication and after the belt is thrown off will frequently run on for two or three minutes before stopping. At the same time when standing still it is difficult to slide the spindle through its bearings so close is the contact."

[1] For a discussion of air lubrication see paper of Professor Kingsburry, "Experiments with an Air-lubricated Journal," published in the *Journal of the American Society of Naval Engineers*, Vol. IX, No. 2, 1897.

SECTION VIII

Three Important Bearing Inventions

Before taking up typical designs and construction for journal bearings, three important inventions must be presented. These three more than all the others have influenced bearing development not only in this country but also abroad. The reason for presenting them is all the stronger because in regard to at least two of them, there is a general misapprehension as to the real inventor.

Taken in chronological order, they are the process of babbitting, the ball and socket seat, and ring oiling.

The fame of Isaac Babbitt is generally supposed to rest upon his discovery of the white metal alloy called to-day babbitt metal. As a matter of fact his invention of the process of "babbitting" has had a much wider influence on the construction of machinery bearings. While there are numberless lining

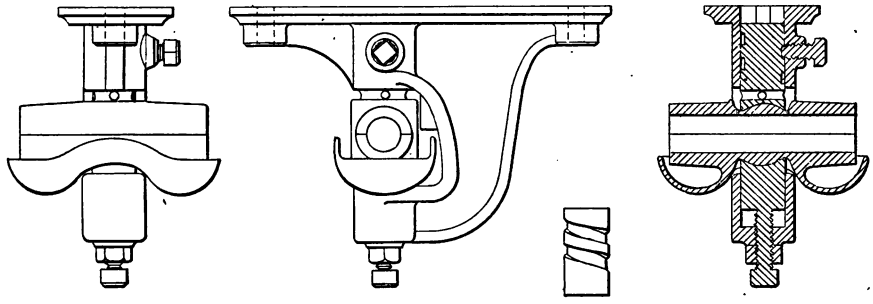


FIG. 30. REPRODUCTION OF DRAWING OF BALL-AND-SOCKET BEARING SEAT ACCOMPANYING BANCROFT OR SELLERS PATENT.

metals, many of which are far different from his original formula, the majority of machinery bearings made to-day consist of a soft metal lining anchored by recesses into a shell.

In a little booklet issued by the inventor in 1848 he says: "Isaac Babbitt would inform the public that his patent does not consist in the *use* of soft metal, simply, but *in the mode of its application and confinement* in boxes prepared for the purpose."

Babbitt's patent was No. 1252, issued July 13, 1839. It was apparently very profitable. The Baltimore & Susquehanna Railroad Company paid him $1,050 for the right to use it on their machinery, including locomotives and cars.

On September 20, 1842, the secretary of the United States Navy, acting under authority of Congress, paid him $20,000 for a license to use the patent in the construction of bearings for the machinery of war vessels.

The second of these great inventions is the so-called Sellers' bearing, invented by Edward Bancroft. Fig. 30 is reproduced from the patent. The following is taken from a letter written by Coleman Sellers, Jr.:

"The Bancroft patent for this type of hanger was originally granted October 9, 1849. After the death of Mr. Bancroft, it was surrendered and re-issued May 12, 1857, the re-issue number is 463.

"Mr. Bancroft was in partnership with the late William Sellers under the firm name of Bancroft & Sellers. They were manufacturers of shafting and mill gearing, and also makers of lathes, planers and other machine tools. This patent covers the idea of forming a portion of each box of spherical form and supporting them in guiding sockets so 'as to allow the boxes thus held to play within the sockets in the manner of a universal joint substantially as described.'

"Mr. Bancroft took out another patent May 22, 1849, in which the box was supported on trunion screws in a vibrating yoke."

The third of there inventions is that of ring oiling. The inventor is John E. Sweet, and the device was never patented, although it is universally used to-day in the bearings of high-grade machinery, particularly electric motors and generators.

Professor Sweet writes as follows in regard to its early development:

"The oil ring was first shown and described in *Engineering* (London) for January, 1868 page 44. The description is this: 'A self-lubricating journal box is shown in which it is pro posed to return the oil from the drip cup to the shaft by means of two loose rings.'

"About all that I can say is—that when I returned from England, I went into the machine shop at the Navy Yard and saw on the floor a half dozen or more hangers made from the drawing in *Engineering*, and I learned that they were to be sent to some other Navy Yard, possibly it was Beaufort.

"When I reached home I found my brother was putting up a lineshaft and also using the same thing. When I built the first Straight Line engine and all afterward, it was one of the features, but so far as I know, no one else ever used it until we exhibited an engine at the Mechanics' Fair in New York, when Mr. Edison got on to it and began using them in his dynamos; after that it began to be used by others in various places.

"Had I taken out a patent at first it would have expired before anyone else adopted it, and like many another patent it would not have been worth the paper it was written upon."

It is not uncommon to find bearings using all three of these inventions.

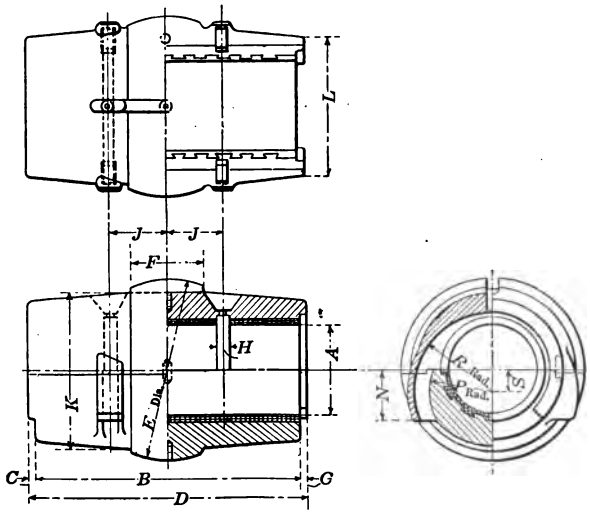
SECTION IX

Typical Designs and Constructions

This section presents, by line drawings and a few halftone engravings, a number of typical bearing designs, constructions and lubricating systems that are representative of American practice. In the space available, it is obviously impossible to present more than a very few such illustrations.

ELECTRIC MOTOR AND GENERATOR BEARINGS

Figs. 31 to 37, inclusive, show several types of electrical generator and motor bearings. Figs. 31, 32, and 33 are from the General Electric Company. The first shows a large spherical-seated, babbitt lined, oil-ring lubricated bear-

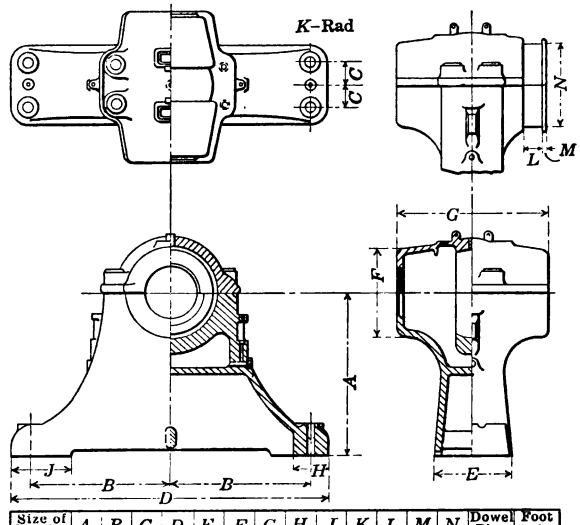


Nominal Dia. of Bearing	Dimensions												
A	*B*	*C*	*D*	*E*	*F*	*H*	*J*	*K*	*L*	*N*	*P*	*R*	*S*
7	21	1/8	22	14 1/16	5 1/8	1	4 1/4	12 1/8	10 3/4	2 1/2	5 3/8	6 5/8	2 3/4
8	24	1/8	25	16 1/16	6 3/8	1 1/4	4 3/8	14	12 1/2	3	6 3/8	7 3/4	2 3/4
9	27	1/8	28	18 1/16	7 1/4	1 1/4	5 1/4	15 3/4	14	3 1/2	7	8 1/8	3

FIG. 31. DESIGN OF 3:1 SPLIT SPHERICAL SEAT BEARINGS.

ing, for nominal diameters of from 7 to 9 in. This type is also used for much smaller bearings. Fig. 32 shows a pedestal box, while Fig. 33 shows a form of 3 to 1 bronze sleeve bearing for small journal diameters.

Figs. 34 to 37, inclusive, are drawn from the practice of the Westinghouse Electric and Manufacturing Company, and need no detailed description. A



Size of Bearing	A	B	C	D	E	F	G	H	J	K	L	M	N	Dowel Pins	Foot Bolts
7″ x 21″	18	20¾	2⅞	47	11½	15	25	5½	9¾	2⅜	3	¾	15	1	1¾–5
	21	21¾	3⅛	49	12										
	24	22¾	3⅜	51	12½										
	27	23¾	3⅝	53	13										
	30	24⅝	3 9/16	55	13⅝										
	33	25⅝	4⅛	57	14¼										
	36	26⅝	4 9/16	59	14⅝										
	39	27⅝	4⅞	61	15⅛										
	42	28¾	5⅛	63	16										
	45	29⅝	5⅜	65	16½										

FIG. 32. DESIGN FOR 3:1 SPHERICAL SEAT PEDESTAL BEARINGS.

feature worth considering in Fig. 37 is the cover of the bearing housing permitting inspection of the oil ring and the condition of the lubrication. This cover is under spring control, is locked by a winged nut and provided with packing to make a dust-proof joint. The oil reservoirs of all these bearings have drainage plugs.

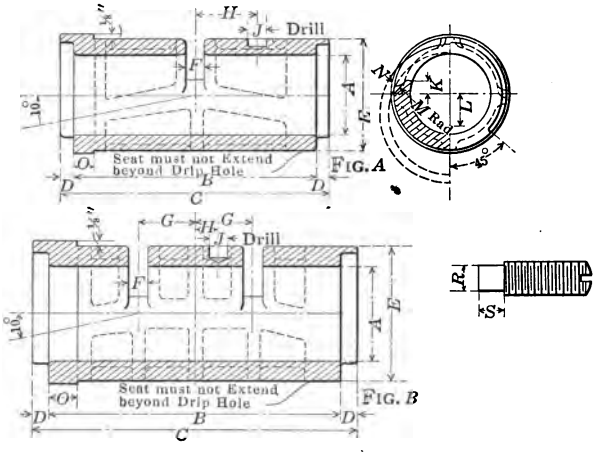
HEAVY GRINDER SPINDLE BEARINGS

Fig. 38 shows a typical construction for heavy grinding spindle bearings from the Norton Grinding Company. The wipers are of felt. In more recent

constructions wood has been substituted, as it is believed that fibers of the felt tend to be picked up and carried into the bearing.

FLOODED LUBRICATION IN MACHINE TOOL PRACTICE

The recent designs of vertical turret lathes built by the Bullard Machine Tool Company have a system of "continuous flow" or flooded lubrication. In the base of the machine is an oil reservoir to which all of the oil drains after passing through the various bearings and gear boxes.



Nominal Dia. of Bearing	FIG.	Dimensions													Dimensions of Set Screw		
A		B	C	D	E	F	G	H	J	K	L	M	N	O	P	R	S
1½																	
1¾	A	5¾	6	3/8	2¾	7/16		1⅛	17/32	¼	11/16	1 7/16	⅜	½	⅝-11	½	3/8
2	A	6	6⅜	3/8	3	7/16		1⅜	17/32	7/16	13/16	1 11/16	⅜	½	⅝-11	½	3/8
2¼	A	6¾	7⅛	3/8	3¼	7/16		1¾	17/32	7/16	⅞	1 13/16	7/16	¼	⅝-11	½	3/8
2½	A	7½	8¼	3/8	3½	9/16		1⅞	21/32	½	1	2 1/16	5/16	11/16	¾-10	⅝	½
2¾	A	8¼	9	3/8	3⅞	9/16		2	21/32	½	1 1/16	2¼	5/16	11/16	¾-10	⅝	½
3	B	9	10	½	4⅛	9/16	1¾	¾	21/32	9/16	1⅛	2 7/16	5/16	⅞	¾-10	⅝	½
3½	B	10½	11½	½	4⅞	9/16	1⅞	¾	21/32	⅝	1¼	2 13/16	3/8	⅞	¾-10	⅝	½
4	B	12	13	½	5⅝	¾	2⅛	1	21/32	11/16	1⅜	3 1/16	¼	⅞	¾-10	⅝	½
4½	B	13½	14½	½	6¼	¾	2⅛	1	25/32	¾	1½	3½	¼	1	1-8	13/16	⅝

FIG. 33. DESIGN FOR 3:1 BRONZE SLEEVE BEARINGS.

Submerged therein and driven from the driving shaft is a geared pump which delivers the oil to a distributing reservoir located on the outside of the column at a height to give a slight head for the free flow of oil through pipes leading to the bearings and gear chambers. The excess oil pumped to the distributing reservoirs drains to the sump through an overflow.

In the pipes leading out of the reservoir are sights so that the flow can be observed, and if a stoppage occurs the overflow from this point at once calls attention to it.

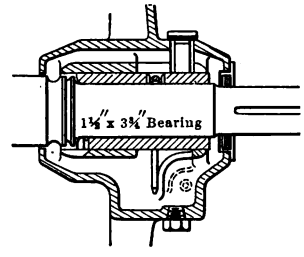


FIG. 34. MOTOR RING OILING BEARING.

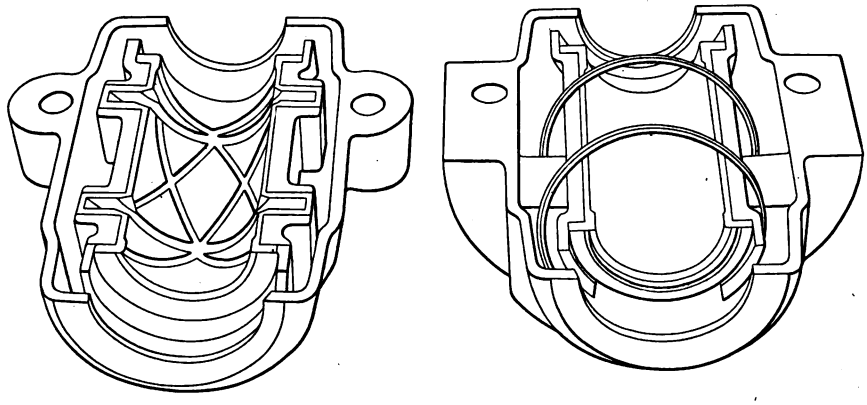


FIG. 35. RING-OILING MOTOR BEARING SHOWING OIL RINGS AND GROOVES.

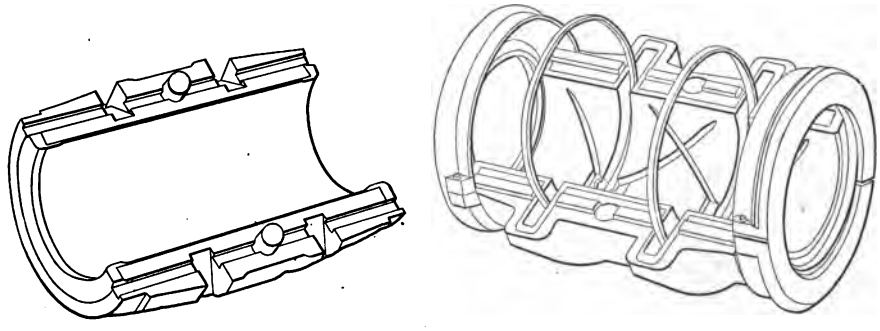


FIG. 36. RING-OILING MOTOR BEARING SHOWING OIL RINGS AND GROOVES.

The flow of oil is at the rate of 0.08 gal. per square inch of projected bearing area per minute.

Fig. 39 shows the table spindle bearing and method of lubricating of one of these machines.

MACHINE TOOL BEARINGS

The illustrations from machine tool practice, Figs. 40 to 48, inclusive, are from the Pratt and Whitney Company and show bearings from a number of different machines.

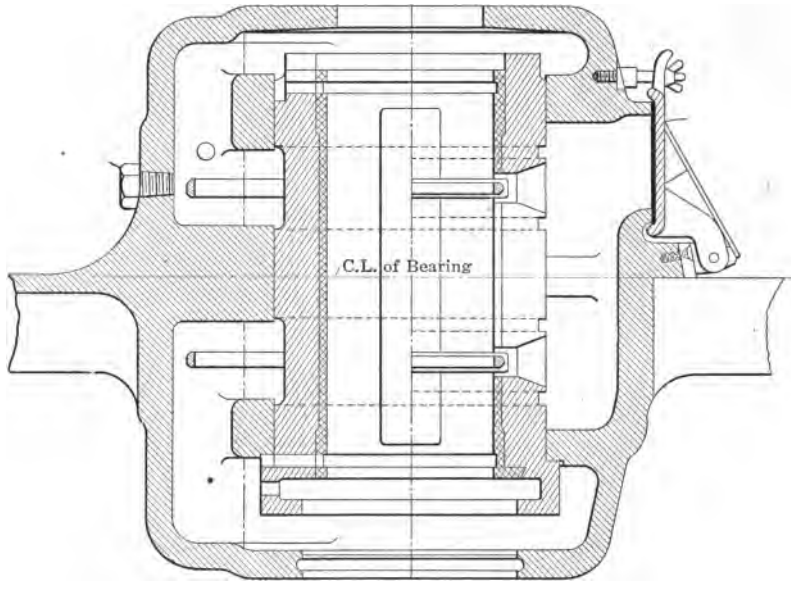


FIG. 37. RING-OILING MOTOR BEARING—RIGID SLEEVE TYPE.

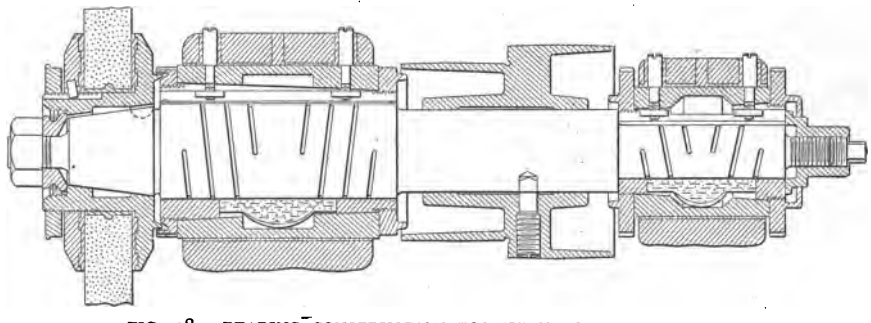


FIG. 38. BEARING CONSTRUCTION FOR HEAVY GRINDER SPINDLE.

Fig. 40 is a section of the front spindle bearing of a turn-table lathe. It is of bronze, tapered, split, with an adjusting nut on each end, an oil hole top and

bottom with a longitudinal groove running through each. These holes connect with an annular chamber in the headstock casting.

Practice limits the rubbing velocity for a bearing of this kind to 800 ft. per minute.

Fig. 41 shows a spindle bearing of a 2 1/2×26 in. turret lathe. It is babbitted with a method of anchoring the babbitt which is used generally on

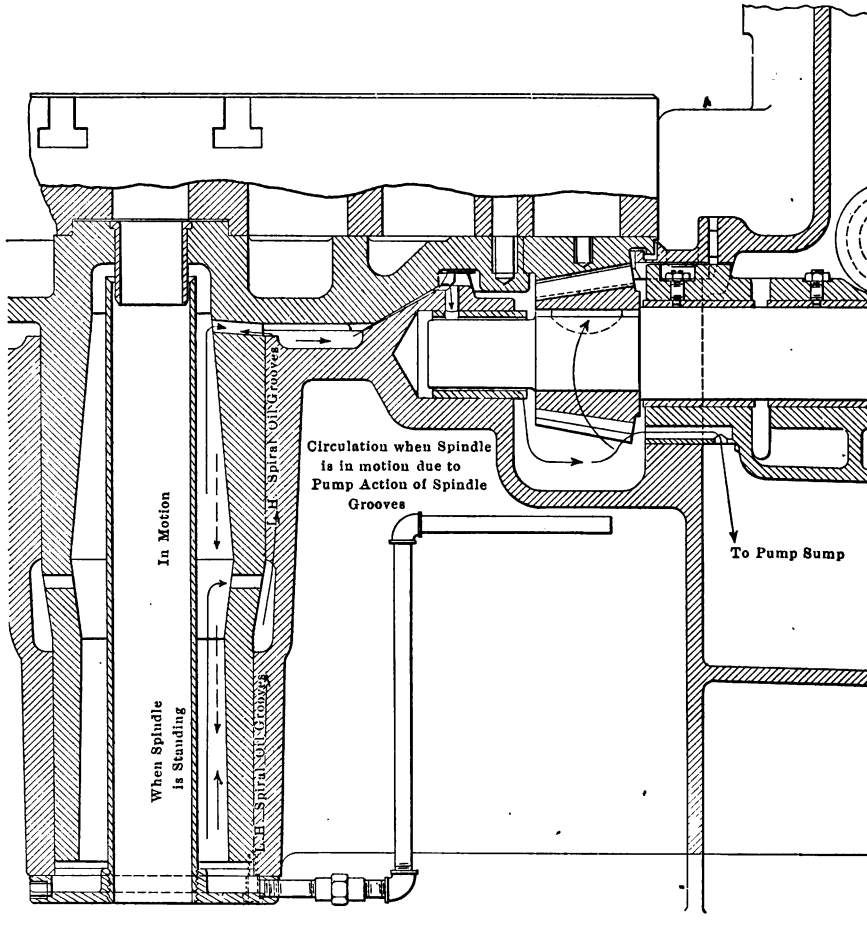


FIG. 39. TABLE SPINDLE BEARING OF VERTICAL TURRET LATHE.

Pratt and Whitney tools. This consists in turning a dovetailed groove at each end of the bearing housing and then drilling a number of holes starting in the surfaces where the bearing is split in such a manner that there holes break through into the lining space at the joining surface of the cap and bottom

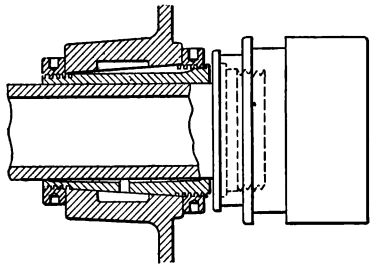


FIG. 40. FRONT SPINDLE BEARING OF TURN-TABLE LATHE.

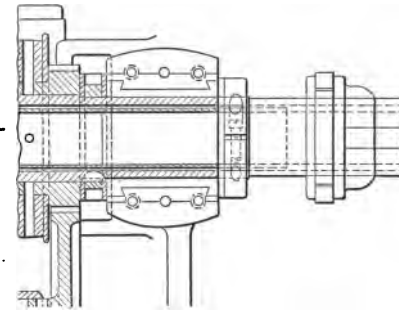


FIG. 41. FRONT SPINDLE BEARING OF TURRET LATHE.

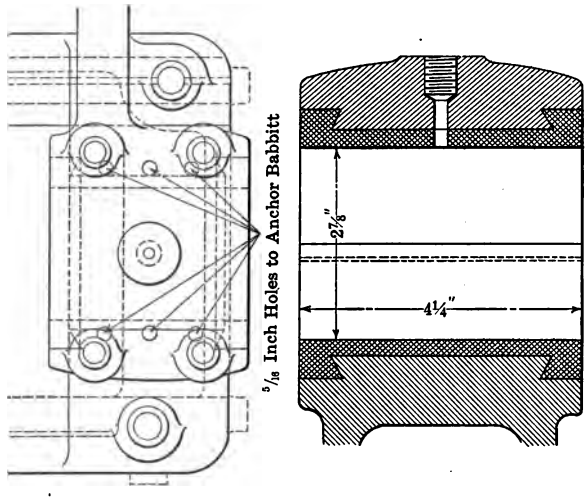


FIG. 42. DETAIL OF ANCHORAGE FOR BABBITT LINING.

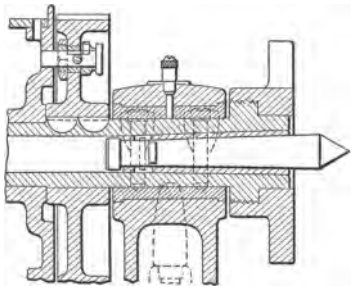


FIG. 43. FRONT SPINDLE BEARING OF 14-INCH TOOL-MAKER'S LATHE.

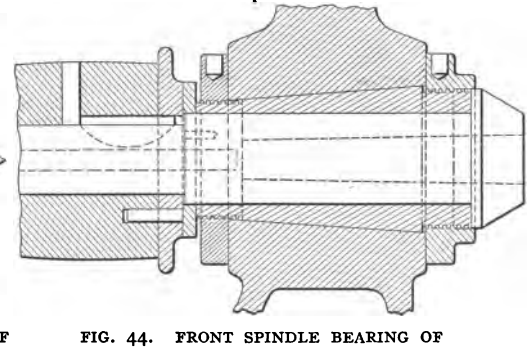


FIG. 44. FRONT SPINDLE BEARING OF HAND MILLER.

half of the box. The babbitt enters these holes and is thereby held securely at the edges of the split, at the same time due to its contraction, it is drawn tightly into the dovetailed grooves at the ends of the box, see Fig. 42.

The front spindle bearing of a 14 in. tool-maker's lathe is shown in section in Fig. 43. It is babbitted with the regular method of anchorage. The proportions are a little less than 1:2, the dimensions being 2 7/16×4 in.

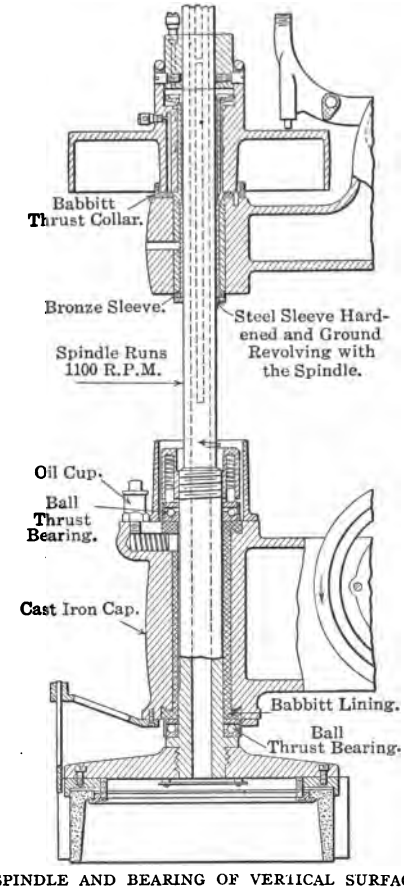


FIG. 45. SPINDLE AND BEARING OF VERTICAL SURFACE GRINDER.

Fig. 44 shows the construction of the front spindle bearing of a No. 10 hand miller. This is of bronze with the same device for taking up wear as in Fig. 40. The bearings are also made babbitted.

The spindle and bearings of a vertical surface grinder are shown in Fig. 45. Here the lower bearing is babbitt lined with ball thrust collars, and the upper is also babbitt lined with a babbitt thrust collar. Instead of anchoring by

drilling a number of small holes at the split surfaces, a number of large holes are drilled through the cap and its mating half. These serve as anchorage with the dovetailed grooves at the ends.

The front spindle bearing of a 5×48 in. cylindrical grinder is shown in

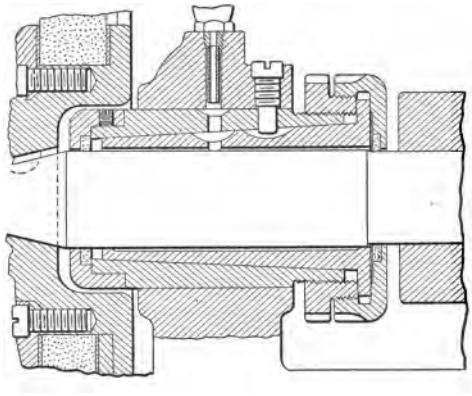


FIG. 46. FRONT SPINDLE BEARING OF 5 x 48-INCH CYLINDRICAL GRINDER.

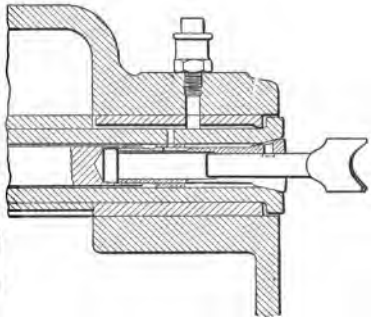


FIG. 47. FRONT SPINDLE BEARING OF SPLINE MILLER.

section in Fig. 46. It is a bronze sleeve with oil grooves top and bottom. In the driving shaft bearings of this machine felt plugs are inserted in the bottom of the sleeve and the Osgood system of grooving is used.

The front spindle bearing of a spline miller is shown in Fig. 47. It is of

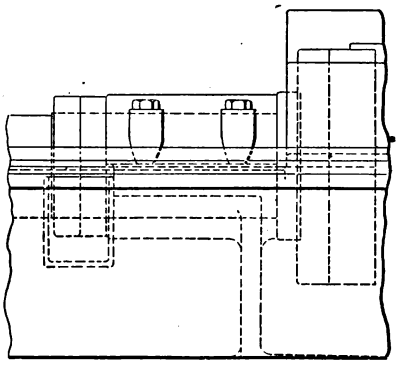


FIG. 48. SPINDLE BEARING OF CUTTING-OFF MACHINE.

hardened and ground tool steel. The rear bearing is bronze. The spindle is likewise of hardened and ground tool steel with a clearance of 0.0005 in. on the diameter. The housing has a chamber for oil.

A large cast-iron spindle bearing of a cutting-off machine is shown in

Fig. 48, and is of interest as being an example of a cast-iron spindle running in a cast-iron box. The speed is slow.

BEARINGS FOR STEAM TURBINES

The step bearings of vertical Curtis turbines are the most prominent example of the use of a cast-iron step for heavy pressures and high speeds. Table 36 page 120 gives the range of pressures, speeds, and quantities of oil used. The bearing plate, or lower block, is of cast-iron rigidly held by the frame. See Fig. 49. The block is guided at the sides and carried on a large screw, passing through a steel nut and coming in contact with a steel block set in the

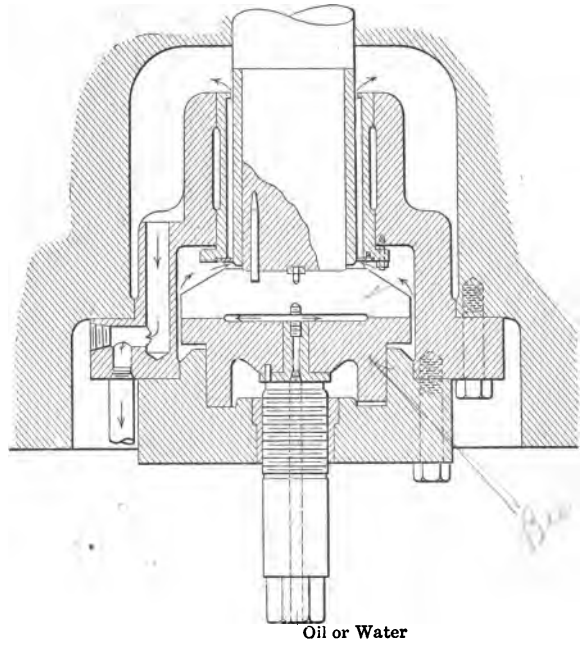


FIG. 49. STEP BEARING FOR VERTICAL CURTIS TURBINE.

bearing plate. It is essential that this plate should be rigidly held, that its upper face should be a true plane set at right angle to the shaft axis, and that the clearance should be so small that the relative alinement of blocks and shaft cannot vary appreciably.

The step plate is likewise of cast-iron and keyed to the lower end of the shaft. Both plates are recessed so that the surfaces of contact are collars. Directly above the step plate is a cylindrical guide bearing. The contact faces of the blocks must be truly parallel, and the contact surfaces of the end of the

screw beneath the lower block and its mate must be likewise true and free from convexity.

Oil is introduced through the center of the screw, passes upward, enters the recess in the center of the plate, passes out between the contact surfaces, and ascends upward through the guide bearing, as indicated by the arrows in the illustration.

These bearings can be run with either oil or water as the lubricant. In service the plates are actually separated by a lubricating film; some four or five times as much lubricant as is necessary is usually pumped as a safeguard. The greater the quantity the greater the separation between the plates and the

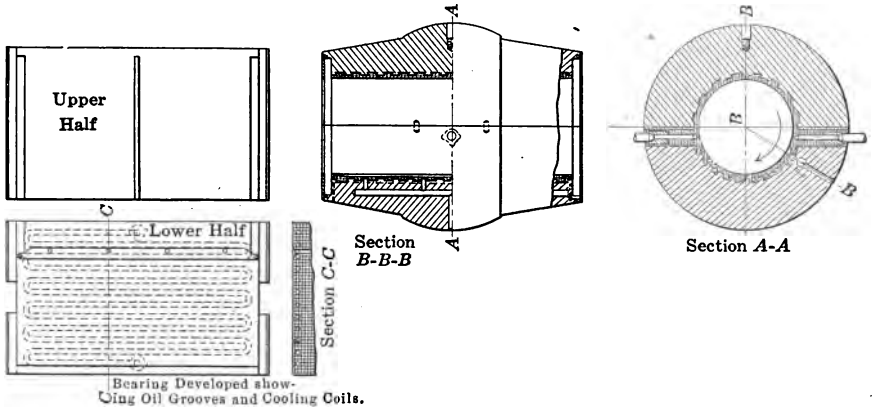


FIG. 50. WATER-COOLED BEARING FOR HORIZONTAL TURBINE.

less the danger of cutting out. The actual separation of the plate is, of course, only a few thousandths of an inch. From this fact it can be seen that this type of bearing calls for the best of workmanship.

Fig. 50 shows a water-cooled bearing for Curtiss horizontal turbines including the arrangement of the cooling pipes and the oil grooving. The bearing is provided with a coil of thin copper tubing cast into the babbitt lining as close to the contact surface as possible, and having at the ends steel blocks securely brazed on. Water is led in through pipes passing through stuffing boxes in the bearing standard. In this way the heat of the bearing is taken up by the cooling water at the point where it is generated. There are no cooling coils in the oil tanks.

The top half, or cap of the bearing, is cut away, except at the ends and center. This relieving is about 0.020 in., so that the oil fed to the bearing fills the space between the shaft and lining.

The oil grooving in the lower half of the bearing is one straight groove into which oil is fed through a central hole as indicated in the cut, or, from several

holes leading to a header hole, which in turn is fed from the pipe in the standard. This groove is well rounded over on the side toward which the shaft rotates; on the side toward the bottom of the bearing it is gradually tapered up to the bearing surface. This assists the action of the shaft in pulling oil into the region of greatest pressure.

The ends of the main oil groove terminate in very small grooves which run through to the ends of the bearing. This allows for the washing out of small particles of grit and dirt, instead of letting them remain at the ends of the bearing and cut the shaft at that point.

The oil is discharged from both ends around the entire circumference and particularly at the point where the ends and middle bands on the upper half of the bearing are cut away.

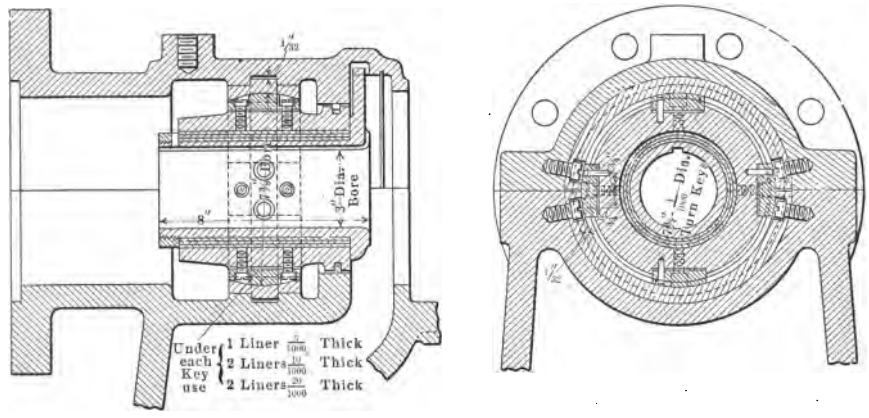


FIG. 51. FLOATING SLEEVE TYPE OF HORIZONTAL TURBINE BEARING.

At the ends of the bearing the casting is allowed to project slightly and a sheet brass plate is attached running down to a small clearance around the shaft. In this recess so formed a deflector or fan is provided, which throws off the oil from the ends of the bearings, preventing it from leaking out at the ends of the standard and from being slooped over the inside of the standard and finally seeping out of the joints.

Bearings of this type are designed to run with an oil pressure of from 15 to 25 per square inch. The water supply pipes are provided with stop cocks to regulate the amount delivered and the discharge is open. Water under low pressure as generally supplied from city mains is used. Most bearings of this kind are arranged so that the temperature of the oil can be taken and the amount of circulating water varied to meet the conditions of running.

Fig. 51 shows a type of floating sleeve bearing used on smaller Westinghouse-Parsons turbines running at 3600 revolutions per minute. The ratio of diam-

eter to length runs from 1:2 1/2 to 1:3. The bearing proper consists of a bronze sleeve provided with an oil groove at the top and prevented from turning by a tail projecting upward into a recess in the cap. Around this tube are three concentric sleeves, each diametral clearance being from 0.004 to 0.006 in. Each has four axial rows of holes to distribute oil and thus maintain a film between each pair of mating surfaces. This construction cushions the shaft and allows it to rotate around its gravity axis rather than its geometric axis if the two are not coincident.

The sleeves are held in place by a nut which in turn is checked by a small set screw. The tube with its nest of sleeves is held in a cast-iron shell having

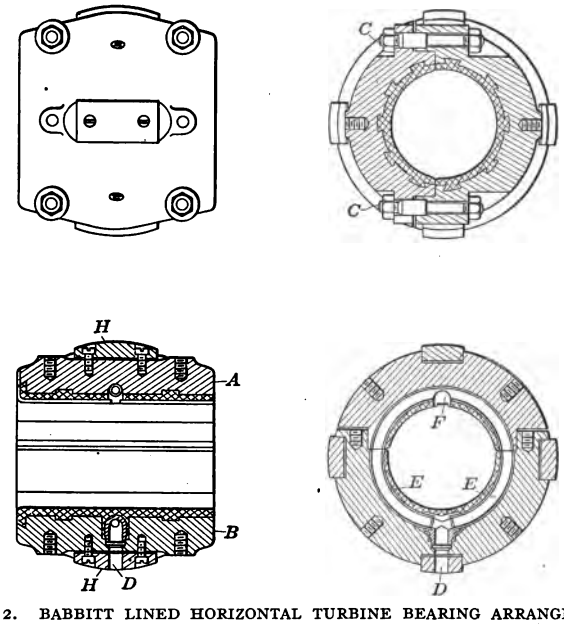


FIG. 52. BABBITT LINED HORIZONTAL TURBINE BEARING ARRANGED FOR FLOODED LUBRICATION.

steel blocks resting in a cast-iron spherical seating ring in the pedestal. Under the blocks are thin liners used to aline the bearing. The adjustment is in increments of 0.005 inch.

Oil under a slight pump pressure, 2 to 5 lb., enters the bearing at the end, passes through it, and drains into a well in the pedestal.

In some cases the bearing tube has been lined with babbitt.

In the larger machines running at 1800 revolutions per minute or less a cast-iron shell, babbitt lined bearing is used without any sleeves. The ratio of diameter to length is from 1:2 to 1:2 1/2. See Fig. 52.

It is split longitudinally in the usual manner and is provided with spherical seating blocks as described above.

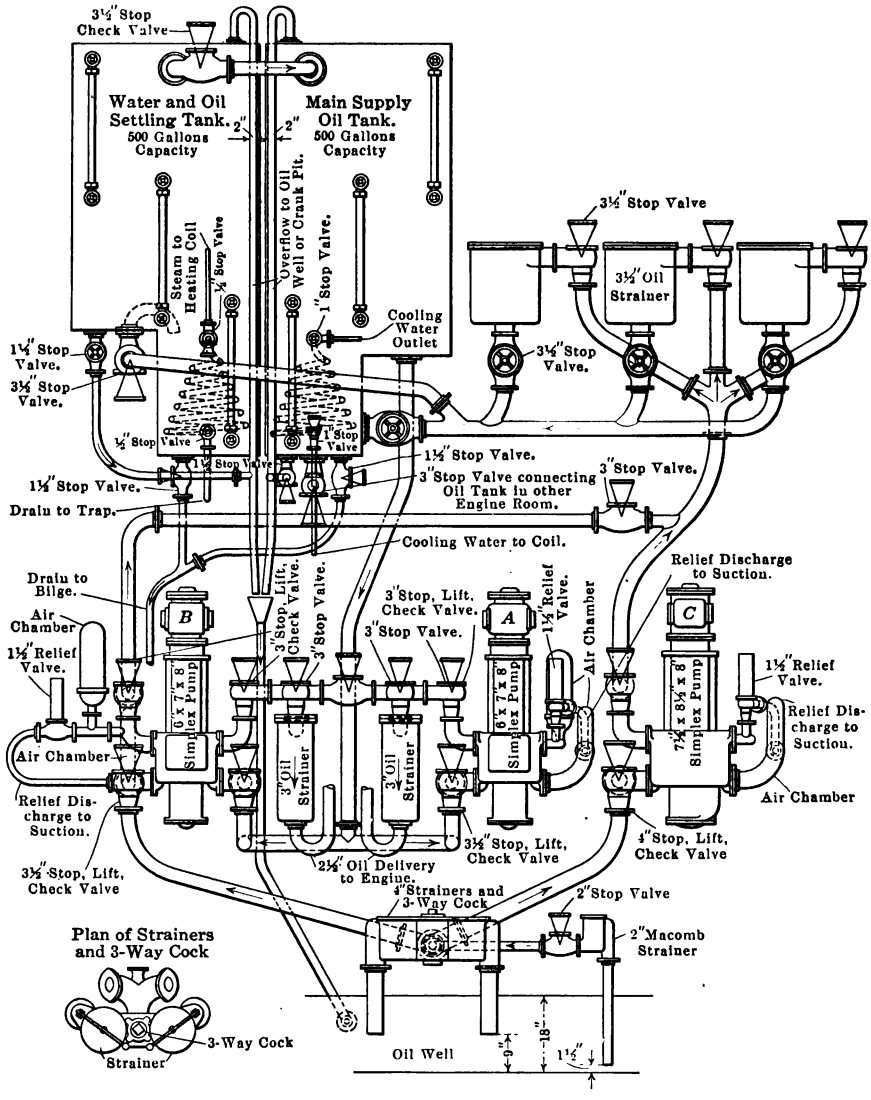


FIG 53. TYPE PLAN OF FORCED LUBRICATION FOR RECIPROCATING ENGINES OF UNITED STATES BATTLESHIPS AND ARMORED CRUISERS.

At the bottom oil enters a copper tube that is set in a recess in the shell and held by the lining; it divides into two streams, passes around the bearing and

into a longitudinal groove in the top of the cap cut to within about 3/8 in. of the ends. It escapes at the ends of the bearing and drains to a well in the pedestal. The babbitt is bored out so that the horizontal dimension of the bore is greater than the vertical and thus greater than the shaft diameter. The fit is over about two-thirds of the bottom half of the circumference. This

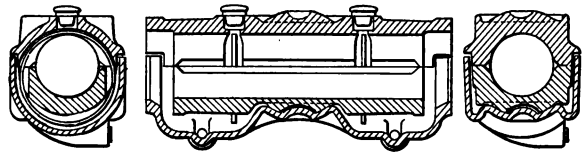


FIG. 54. BALL-AND-SOCKET RING-OILING LINE SHAFT BEARING.

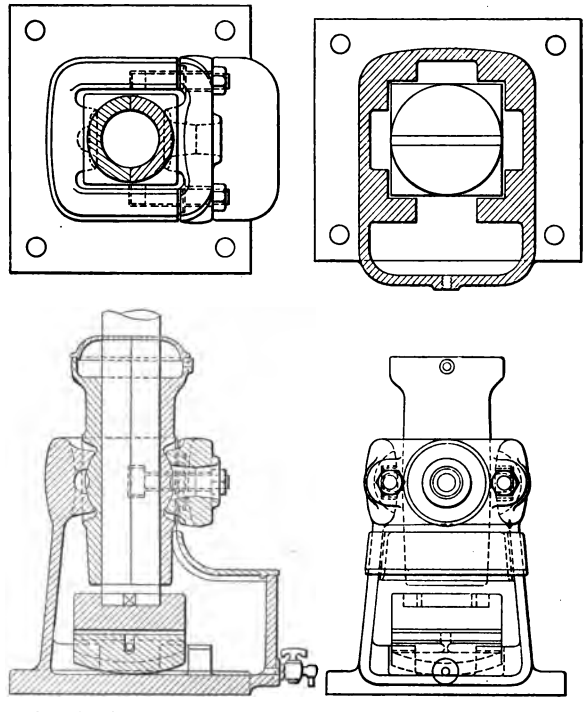


FIG. 55. STEP AND GUIDE BEARINGS FOR VERTICAL SHAFT.

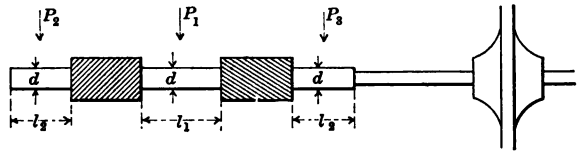
prevents all side binding on the journal, does away with any edges to scrape off the oil and provides a small reservoir of oil within the bearing.

Flooded lubrication is used at low pressures as in the other type. The oil is cooled by passing it through pipes submerged in water or through a tank containing cooling coils.

At the open ends of the bearing housings are baffles that prevent leakage.

The clearance between the shaft and the knife edges of these thin collars is from 0.020 to 0.030 in.

Albert E. Guy gives both the European and standard practice in regard to the design of De Laval steam turbine bearings in Tables 38 and 39. This is of interset and value because of the extremely high speeds. The bearings are made of brass shells lined with babbitt. The pinion shaft bearings are lubricated by sight feed, or wick feed. The power shaft bearings are ring oiled with one ring in the middle of each bearing.



H. P.	R.P.M.	d	l_1	$\frac{l_1}{d}$	p_1	A_1	l_2	$\frac{l_2}{d}$	p_2	A_2	V
											ft. per sec.
	20700	0.788″	4.56″	5.75	17.8	1.73	3.94″	5.	13.36	1.3	71.1
30											
55	16600	1.102″	5.91″	5.36	19.25	2.10	5.67″	5.15	13.36	1.45	79.7
75	16700	1.181″	8.27″	7.	17.64	2.05	7.29″	6.17	13.36	1.56	85.9
110	13000	1.575″	9.05″	5.75	17.8	2.17	8.11″	5.15	13.36	1.62	89.2
150	13000	1.575″	11.42″	7.25	19.2	2.34	10.24″	6.5	14.22	1.85	89.2
225	11000	1.968″	12.21″	6.2	19.2	2.47	10.83″	5.5	14.22	1.85	94.5
300	10600	2.165″	14.57″	6.72	19.2	2.63	13.11″	6.05	14.22	1.94	100.4
450	10600	2.165	21.66	10.	19.35	2.64	17.91″	8.27	15.65	2.13	100.4

d = diameter of journal in inches.

l = length of journal in inches.

p_1 and p_2 = pressure per square inch of projection dl.

A_1 and A_2 = coefficients

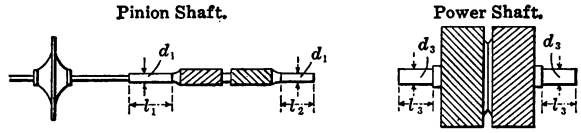
V = peripheral velocity of journal in feet per second. When V = 90 ft. per second forced lubrication must be used—[oil brought to bearing under pressure].

P_1 and P_2 = total pressure on bearing in lb. = $p \times d \times l$.

Load on middle bearing assumed to be 1 1/2 times that on either of the others.

$A = \frac{P \times R.P.M}{168{,}000 \times l}$. A should ordinarily not exceed 2.

TABLE 38.—EUROPEAN BEARING PRACTICE WITH DELAVAL SINGLE GEAR TURBINE: ONE PINION SHAFT, ONE POWER SHAFT.



H. P.	d_1	l_1	l_2	$\frac{l_1}{d}$	$\frac{l_2}{d}$	R.P.M.	Peripheral velocity		d_3	l_3	$\frac{l_3}{d_3}$	R.P.M.	Peripheral velocity
							ft. per sec.						ft. per sec
7	0.395″	$3\frac{3}{8}$″	$2\frac{27}{32}$″	8.55	7.2	30000	51.7	One pinion shaft and one power shaft.	1″	$3\frac{1}{16}$″	3.06	3000	13.1
10	0.395″	$3\frac{3}{8}$″	$2\frac{27}{32}$″	8.55	7.2	24000	41.35		1″	$3\frac{1}{16}$″	3.06	2400	10.475
15	0.532″	$4\frac{1}{8}$″	$3\frac{21}{32}$″	7.75	6.85	24000	55.7		1.3″	$3\frac{13}{16}$″	2.936	2400	13.62
20	0.708″	$4\frac{11}{16}$″	$4\frac{1}{4}$″	6.62	6	20000	61.8		1.31″	$4\frac{1}{2}$″	2.435	2000	11.43
30	0.708″	$5\frac{1}{2}$″	5″	7.77	7.06	20000	61.8		1.46″	$5\frac{3}{16}$″	3.555	2000	12.75
55	0.871″	$6\frac{7}{8}$″	$6\frac{7}{8}$″	7.89	7.89	20000	76		2″	$6\frac{1}{2}$″	4.25	1500	13.1
75	1″	$5\frac{1}{2}$″	$4\frac{7}{8}$″	5.5	4.875	16400	71.5	One pinion shaft and two power shafts.	$1\frac{3}{4}$″	6″	3.43	1500	11.45
110	1.25″	$7\frac{1}{2}$″	$5\frac{3}{8}$″	6	4.3	13000	70.9		$2\frac{1}{2}$″	6″	2.4	1200	13.1
150	1.25″	$7\frac{1}{2}$″	$5\frac{3}{8}$″	6	4.3	13000	70.9		$2\frac{1}{2}$″	$6\frac{1}{2}$″	2.6	1200	13.1
225	1.6″	8″	$5\frac{7}{8}$″	5	3.675	11060	77.2		3″	8″	2.667	900	11.78
300	1.6″	$9\frac{7}{8}$″	$7\frac{1}{4}$″	6.17	4.53	10500	73.3		$3\frac{1}{2}$	$10\frac{1}{2}$″	3	900	13.74

p = pressure per square inch on diametrical plane of bearing.

p = 7 to 15 pounds per square inch up to 55 h. p.
= 15 to 23 pounds per square inch from 55 to 110. h. p.
= 23 pounds per square inch from 110 to 300 h. p
} for pinion bearings.

p = 15 to 40 pounds on power shaft bearings.

Spiral grooves in bearing to distribute the oil are 1/16 inch wide and 3/64 inch deep, with holes at the ends to watch the conditions of lubrication.

TABLE 39.—STANDARD BEARING PRACTICE OF DELAVAL STEAM TURBINES.

FORCED LUBRICATION FOR HORIZONTAL ENGINES

Fig. 53 is a type plan of forced lubrication for reciprocating engines of battleships and armored cruisers as worked out by the Bureau of Steam Engineering of the United States Navy. The location and arrangement of tanks, strainers, valves, and piping is shown.

BALL-AND-SOCKET SHAFT BEARINGS

Fig. 54 shows the ring-oiling line shafting box, and Fig. 55 a usual form of step bearing for vertical shafts as made by Wm. Sellers & Company, Inc. The horizontal box may be oiled by a syphon oil cup through the center or

oil may be placed in the reservoir around the top ball and allowed to find its way through the small holes to the shaft. It is customary to fill the grease cups with tallow or other grease, which will melt if the box becomes at all warm.

LOCOMOTIVE AND CAR BEARINGS

The eleven illustrations, Figs. 56 to 66, inclusive, are reproduced from designs of the Baldwin Locomotive Works. Two are of locomotive driving

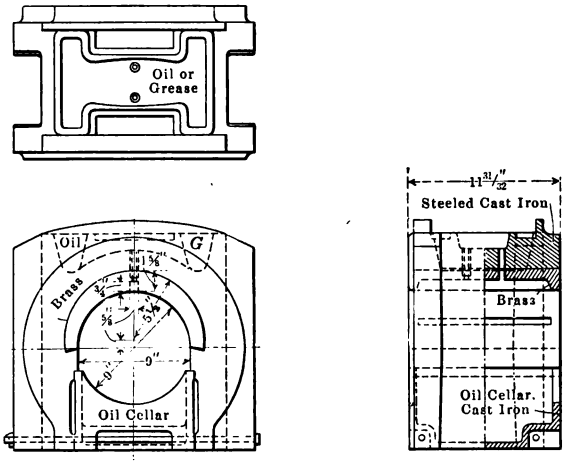


FIG. 56. LOCOMOTIVE DRIVING BOX.

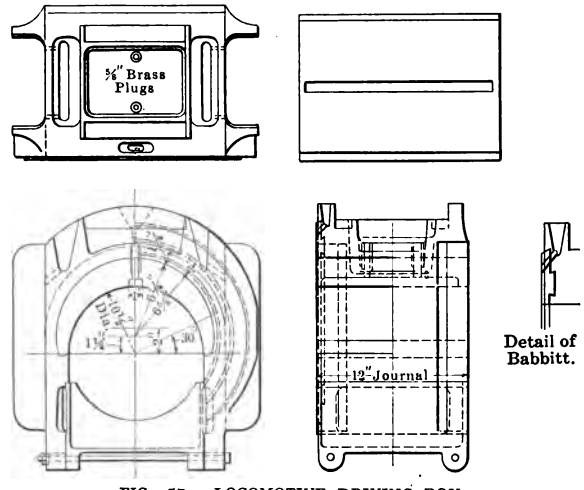


FIG. 57. LOCOMOTIVE DRIVING BOX.

boxes, two of tender boxes, one of an engine truck box, one of a crosshead, one of an eccentric strap and four of rod stubs. A few dimensions are included in each case to give an idea of proportions.

As a matter of interest, Fig. 67 is reproduced. It is of a ring oiling car box invented and developed by W. O. Dunbar, and at one time experimented with on the Pennsylvania Railroad.

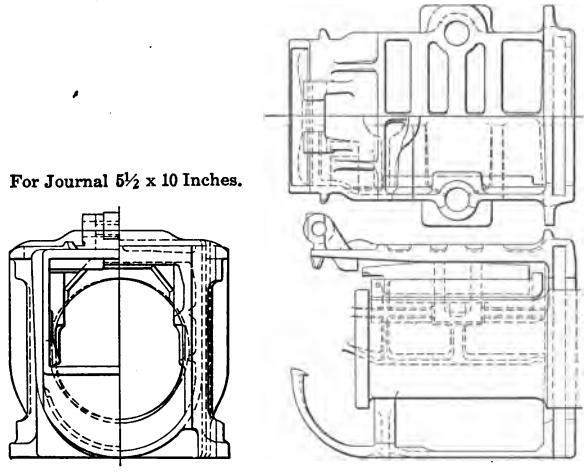


FIG. 58. LOCOMOTIVE TENDER BOX.

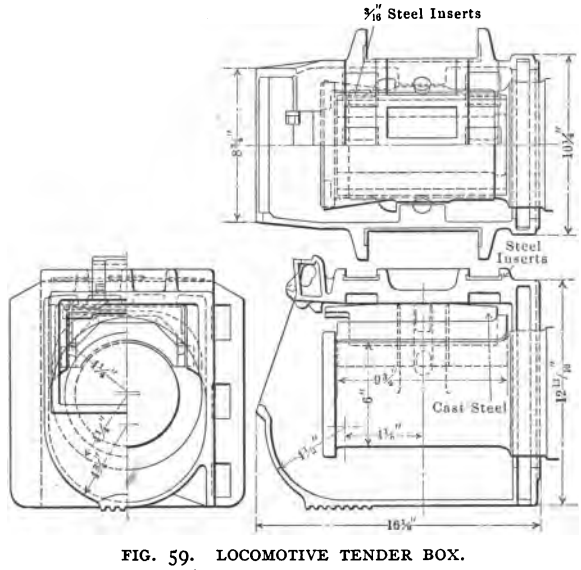


FIG. 59. LOCOMOTIVE TENDER BOX.

KINGSBURY THRUST BEARING

Fig. 68 shows in partial elevation and section, a thrust bearing developed and patented by Professor Kingsbury.

The design differs from the ordinary collar bearing in that one of the pair of collars is divided into several sectors or "shoes" which are individually supported on spherical seats whereby each shoe is enabled to adjust itself to

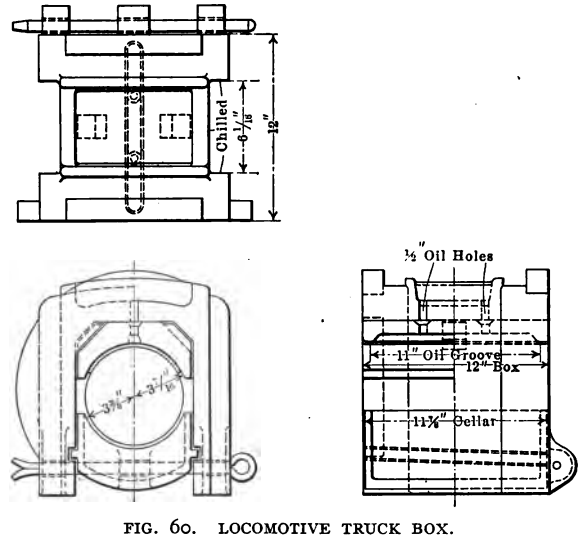


FIG. 60. LOCOMOTIVE TRUCK BOX.

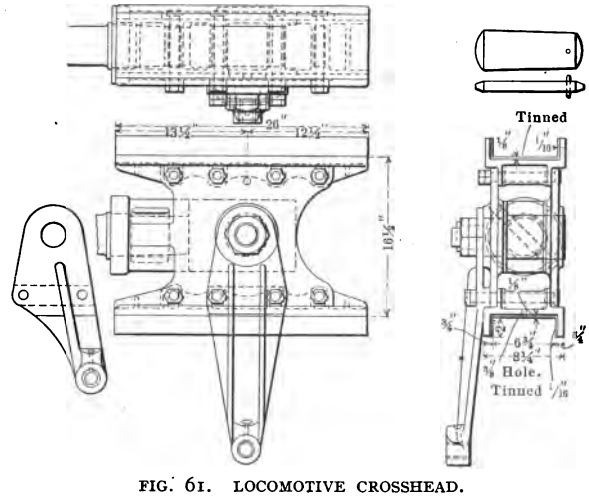


FIG. 61. LOCOMOTIVE CROSSHEAD.

the best alinement and pressure distribution at the bearing face. The concave seats for the shoes are formed in a ring which has a spherical seat in the housing, for maintaining group alinement of the shoes. Lubrication is effected by a

bath of oil in the case of vertical shafts, and by flood lubrication with horizontal shafts.

The considerations leading to the adoption of the construction are as follows: Both theory and experiment have shown that in shaft journals in which perfect lubrication is maintained, the film of oil is invariably wedge-shaped, being

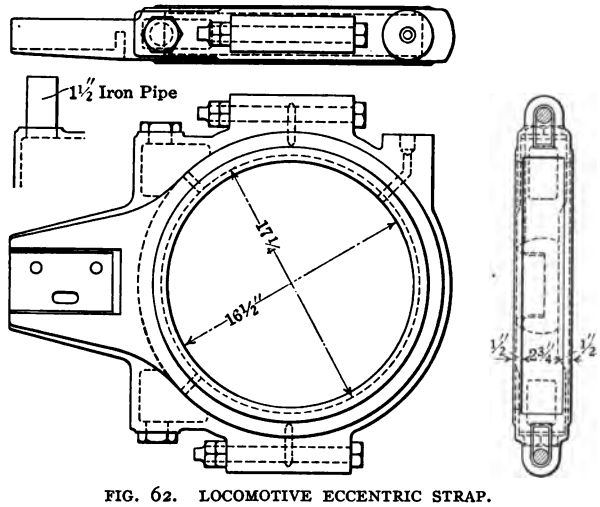


FIG. 62. LOCOMOTIVE ECCENTRIC STRAP.

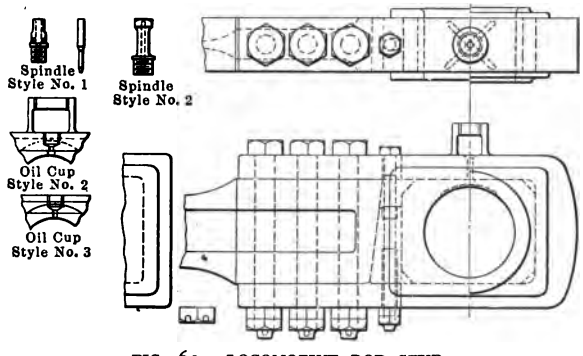


FIG. 63. LOCOMOTIVE ROD STUB.

thicker at the edge where the oil enters and thinner toward the leaving edge. In the ordinary collar bearing, the bearing surfaces remain strictly parallel, hence no wedge-shaped film of oil can form, and even with bath lubrication the action of the lubricant is imperfect, thus the specific loads are necessarily relatively low and the coefficients of friction high. The shoe construction on

the other hand enables the wedge-shaped films to form and thus perfect lubrication is maintained. The leading edge of each shoe is depressed by the entering oil, giving automatically the proper slight inclination between the bearing

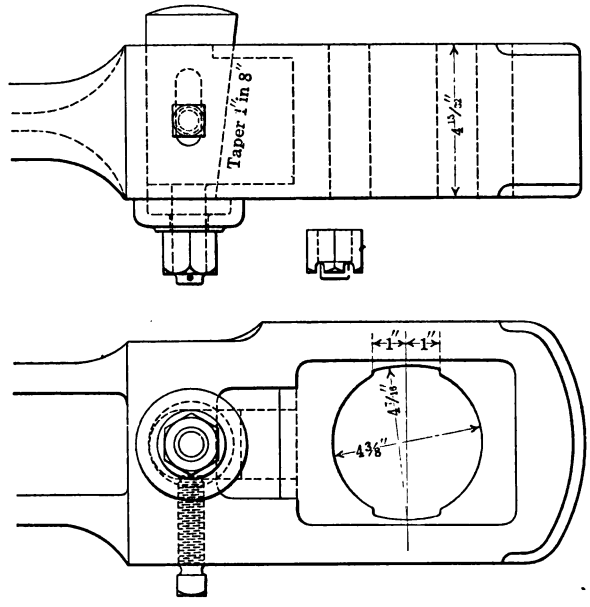


FIG. 64. LOCOMOTIVE ROD STUB.

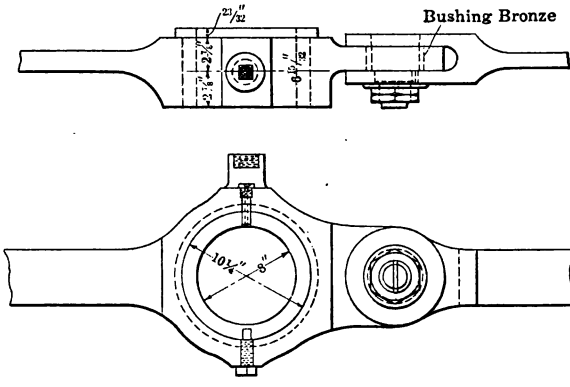


FIG. 65. LOCOMOTIVE ROD STUB.

surfaces. That this actually takes place is shown by the fact that such wear as occurs on the shoes is first indicated at the leaving edge.

The shoe type bearings have shown good results under tests and in service.

Pressures from 250 to 1000 lb. per square inch are borne under ordinary conditions with coefficients of friction as low as 0.0022. Test runs have been made

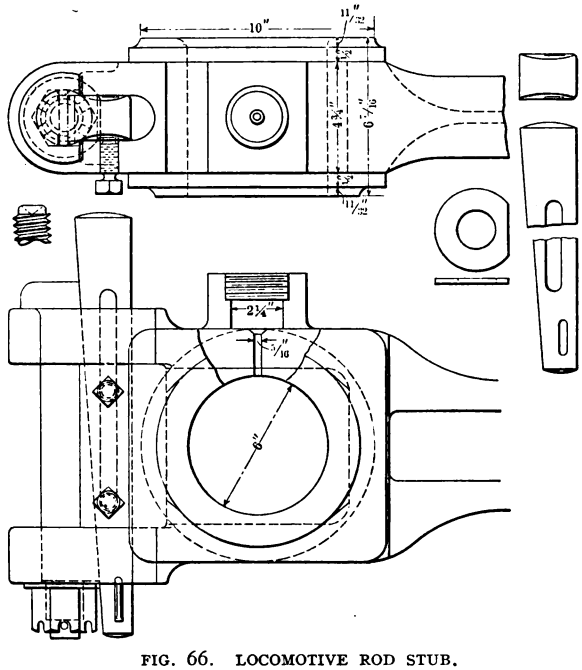


FIG. 66. LOCOMOTIVE ROD STUB.

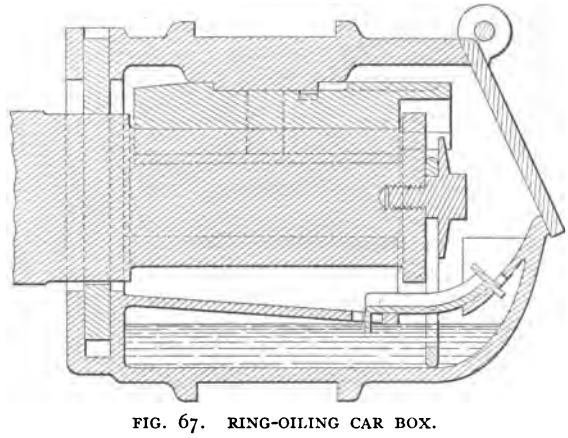


FIG. 67. RING-OILING CAR BOX.

at a circumferential speed of 4500 ft. per minute with pressures up to 2600 lb. per square inch, without causing failure of the oil film; and at lower speeds a pressure of 5000 lb. has been reached experimentally.

TEXTILE MACHINERY BEARINGS

W. S. Southwick sums up American practice in the design and application of the bearings of textile machinery, thus:

"Among the kinds of bearings generally used on textile machinery, there are two or three of interest. The bearing most generally used is a simple cast-iron box mating a steel journal. Many of the caps for larger sized bearings have a recess cast in their upper side to form an oil

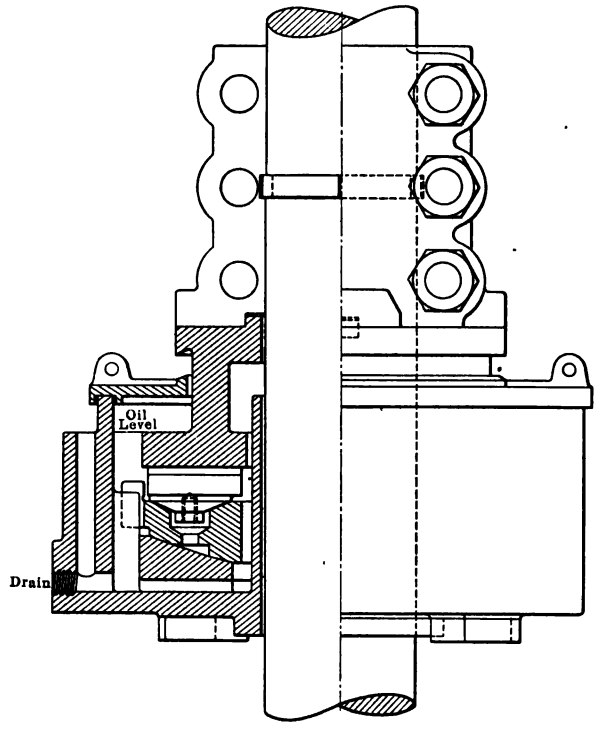


FIG. 68. KINGSBURY THRUST BEARING.

well; the smaller bearings simply have a drilled and countersunk oil hole. A good grade o machinery oil is used for lubrication supplied by a squirt can.

"There are no commonly accepted factors for design. In most cases the design is governed largely by experience and varies with conditions. A device that has been developed to a very high state of perfection, from a mechanical standpoint, is the spindle. Over 500 patents have been issued on it. As now built, it runs at from 8000 to 12,000 revolutions per minute very smoothly, with little wear, and requiring but a small amount of power. The blade which is made of hardened steel, accurately ground to a slight taper, runs in a cast-iron bolster partly covered with cotton packing to reduce vibration. In some constructions the bolster is adjustable to compensate for wear. The cast-iron base in which the bolster is fitted has on one side a projection adapted to receive the oil and convey it to the revolving parts which are surrounded by oil. A light clean spindle oil is used, generally supplied by an oil can.

"Large vertical twister rings are often equipped with a self-oiling device. This consists of a wick which lubricates the traveler at a point below its contact with the yarn. The bearings on the parts which operate the harnesses and shuttle-boxes of fancy worsted and woolen looms are malleable iron and steel case-hardened. These bearings are about 3/4 inch in diameter and 5/32 inch wide. They are lubricated with an ordinary oil cup. In a few cases, chilled cast-iron journals and cast-iron boxes are used. They are oiled in the same manner.

The temple roll is an illustration of a slow running bearing where a brass stud forms the journal and a wooden bushing previously boiled in linseed oil the lining shell. This is inserted into the roll. Albany grease is used for lubricating, being placed in the bushing before assembling."

SECTION X

Hints on the Care of Bearings

Never start a new engine or machine without examining the fit of the bearings and knowing it to be right.

Never start up any machine without knowing that its bearings are properly oiled.

A good way to adjust new engine main bearings is to set the boxes up tight and slack back the adjusting screws one face of the head. Then start the engine, watch it carefully and tighten up the screws gradually until the pound is all taken out.

Remember that the clearance in a cold bearing is greater than in the same bearing at running temperature. When warming up the journal grows larger faster than the bore of the bearing.

Remember that after a machine has been idle for some time the oil is probably squeezed out from between the bearing surfaces.

A reasonable slackness for engine bearings from 6 to 12 in. in diameter is about 1/200 in., for crank pin and cross-head pin bearings up to 4 in. in diameter about 1/250 in. These dimensions can easily be approximated by setting the bearing up tight, then slacking back a certain number of faces on the wedge bolt-head depending upon a number of threads and the taper of the wedges.

A bearing set up too slack is nearly as bad as one too tight. Pounding will make a box run hot and is liable to peen the brass or babbitt and destroy the fit.

Don't squirt oil into an oil hole and hope that it will reach the journal See that the hole is open and not filled with dirt.

Don't put the end of the spout of an oil can into an oil hole and pump the bottom unless there is oil in the can.

Squirt oil into the oil holes and not onto the floor—in the latter place it may lubricate your shoes but it does not help the bearings.

In cold weather a little heat will help the oil to run out of your oil can—the top of the steam-chest is a favorite place for this warming up

Don't neglect a bearing because it is under a bench, or in a dark corner, or in some out-of-the-way place. "Out of sight, out of mind" is poor policy to apply to bearings.

Keep an eye on the oil tanks and do not let them get too low. Good oil cannot always be gotten in a hurry and some bearings may suffer.

In general, the best oil to use on bearings is the lightest oil with which they can be run safely.

Kerosene is successfully used as a lubricant for high-speed, hardened steel spindles and bearings, were the pressures are very light.

Don't depend entirely on the sight glass to tell you how much oil is in the well of an important bearing.

When babbitting a box, a piece of paper wrapped around the shaft or mandrel will keep the hot metal from chilling and reduce the amount of subsequent scraping.

If possible it is a good plan to heat a bearing shell before babbitting.

Never pour babbitt into a shell that contains any water or is not perfectly dry. Many serious burns have come from a failure to observe this rule.

If several small bearings of the same size are to be babbitted, it is better to use a babbitting arbor rather than a finished shaft. If the arbor is sprung a little from the heat, no harm is done.

For severe service it is a good plan to peen the sruface of a babbitted box before it is bored or reamed to size.

Don't try to run babbitt in bearings subjected to heavy shock and pound.

Don't forget to cut oil grooves when you rebabbitt a box.

Don't forget to drill the oil hole when you rebabbitt a box.

Don't cut oil grooves through the end of a box. The oil will run out.

In general, oil grooves should be cut at right angles to the direction of motion of the journal.

If oil will not follow the grooves of a tightly fitted bearing, try venting by a scratch with a file or scraper through to the end in order to let out the air.

Be sure that *both* ends of oil tubes are securely fastened so that they cannot jar loose.

Any self-oiling bearing with an oil well should be washed out at intervals. Be sure the well is refilled before the machine is started.

Be sure that there are no points in oil tubes higher than the point where oil is introduced. Oil will not run up hill.

When making new oiling rings be sure that they are smoothly finished and without any corners or edges to catch when in use and stop the rings from revolving.

When putting in new chains for self-oiling bearings, see that they do not drag on the bottom of the oil well. If they do hit the bottom, a hole will be worn through in time, letting out the oil and probably ruining the bearing.

Don't let induction motor bearings wear sufficiently to drop the rotor against the stator. Put in new bearings.

Felt wipers should be taken out at intervals and soaked in kerosene to soften them and remove the glaze.

Don't let a hot box begin to smoke before you attend to it. You should smell it long before you see it.

The rapid heating of a bearing is a danger sign. If a bearing takes two or three hours to reach a temperature uncomfortably hot for the hand, it is probably safe, but if that same temperature is reached in 10 or 15 minutes there is liable to be trouble.

If you have a hot box on a main engine bearing, first loosen it up a little all around and then give it a generous dose of cylinder oil through the feel holes, but don't try to feed it through the ordinary oil channels; they are too small. If this oil will stick to the shaft it is apt to cure the trouble.

Another good remedy is a mixture of graphite and oil—one part of graphite to 10 of oil. But do not feed this until the graphite has been thoroughly wetted. Dry graphite will not help a hot box.

If there is no graphite handy use flowers of sulphur in the same way; or failing in that, try ground talc. These should be mixed with oil in the same proportions, one of the powder to 10 of oil, and the dry part of the mixture must be thoroughly wetted before it is applied.

In aggravated cases a stream of water from a hose will accomplish a good deal in bringing down bearings temperatures, or ice may be used packed around the bearings.

Hot crank pin boxes may be cooled by fixing a hose in such a position that the crank will strike through the running water at about the upper part of its swing. The same scheme may be used with hot cross-head pin boxes.

Small bearings that heat can sometimes be helped by flooding with kerosene or gasoline to wash out any dirt, grit or metal particles that may be in them. However, this method must be used with caution for the lubricating oil is also washed away. A large supply of regular lubricating oil must be at once poured in. If on hand the kerosene should be used in preference to gasoline as the latter cuts the lubricating oil more quickly.

Sometimes a small bearing that heats can be helped by feeding all of the oil that can be run through it—flooding. This tends to bring down the temperature and may wash away some foreign particles.

If a babbitted bearing heats pull it to pieces and look for black spots. Those are the places to scrape.

Bearings can often be run much hotter than is generally thought. A bearing hot enough to blister the hand will run without injury.

All important bearings, even of the self-oiling type, should be examined at frequent intervals. In many shops it is the practice to inspect all motor and generator bearings, at least once each day, and to thoroughly overhaul all of the line shaft bearings either once every six months or once a year, depending

upon the nature of work. This means taking the bearings to pieces, thoroughly washing all parts, cleaning out the oil wells and refilling with fresh oil.

Where large quantities of lubricating oil are used an oil filter will save money.

As far as possible, do not let any oil from your bearings escape either by dripping or through atomizing. This is one of the most difficult points to care for in practice, particularly with high speed machinery. If oil is thrown out in quantities as in large drops or small streams, it is comparatively easy to stop it. Felt wiping rings at the end of the box are usually effectual but the material must not be too porous, or it may act as a sucking medium and instead of being a remedy be another source of trouble.

It is not unusual to find a central station where the walls of the room are covered with a coating of oil and dirt; around the generators and motors the atomized oil gathers particles of dust, settles in the air passages and may eventually shut off the ventilation. It is most difficult to stop this atomizing action. What must be done is to prevent any current of air passing through the bearing itself or across the surface of any body of oil. If this takes place oil will be vaporized and diffused throughout the entire room. The points where oil escapes can usually be determined by pieces of white paper or pieces of sheet steel painted a light color. The oil stains will show. The only remedy is to entirely shut off all air currents so that they cannot reach the oil. At the same time this is often a baffling problem to solve successfully.

PART II.

BEARINGS WITH ROLLING CONTACT

SECTION I

Rolling Friction and Factors of Design for Ball Bearings

The exact nature of rolling friction is not accurately known, but much splendid experimental work has been done with bearings that involve this principle and we have empirical laws to govern designs. However, there is no theoretical investigation giving results that can be applied to ball and roller bearings in the way that the work of Professor Osborne Reynolds can be applied to journal bearings with sliding contact.

A consideration of the properties of ordinary journal bearings—bearings with sliding contacts—and ball and roller bearings presents a series of contrasts.

At the outset we have sliding contact contrasted with rolling contact.

In the journal bearing the friction is governed by the quality of the lubrication. The better the lubrication the less the friction. In ball and roller bearings the reverse is true. It has been demonstrated by experiment that a well designed ball or roller bearing may give a lower coefficient friction when running clean and dry than when lubricated with even a thin oil. This must not be taken to mean that such bearings should be run without oil or grease, for the reverse is necessary.

In perfectly lubricated journal bearings there is no metallic contact between the surfaces. In ball and roller bearings there is always metallic contact.

In journal bearings many different kinds of materials are used, running from wood and soft babbitt to hardened steel. With a few rare exceptions, the only material used for ball and roller bearings is steel, and in general it is hardened.

The coefficient of friction for sliding contact is much higher than for rolling contact. The coefficient of rest and slow motion for sliding contact is very much greater than for motion at ordinary speeds. The coefficient of friction for the beginning of motion with rolling contact is not much, if any, greater than for motion at ordinary speeds. For this reason ball and roller bearings are peculiarly adapted for machinery having a high starting torque.

In journal bearings running at a high rubbing speed the dissipation of heat becomes an important problem. With properly designed and operated ball and roller bearings the amount of energy liberated in heat is so slight that it is a negligible factor in design. Cooling devices are not used.

With journal bearings practice indicates that a limiting speed is soon reached. For illustration, internal grinding fixtures with such bearings are seldom run

over 12,000 to 15,000 revolutions per minute. By contrast, internal grinding spindles mounted on ball bearings are commonly run at speeds of 30,000 revolutions per minute, are reported to have been run commercially at a speed of 100,000 revolutions per minute, and experimentally at a speed of 120,000 revolutions per minute.

For heavy loads supported on vertical step bearings the ordinary step type with sliding contact has frequently been a source of trouble. On the other hand, there are ball and roller thrust bearings in use carrying loads up to 400,000 lb. and in one installation a load of over 2,000,000 lb. is sustained on a plain roller thrust bearing.

FORMULAS FOR UNIT LOADS

The formulas for specific load, or load per unit of carrying element, are as follows:

$$p = \frac{W}{\frac{n}{c} ld} \quad \text{for roller journal bearings.}$$

$$p = \frac{W}{\frac{n}{c} D^2} \quad \text{for radial ball bearings.}$$

$$p = \frac{W}{n} \quad \text{for roller and ball thrust bearings.}$$

In these formulas W = the total load, p = the unit load, or load per carrying element, l = the length of the journal bearings, d = the diamenter of the rollers in a roller bearing, D = the diameter of the balls in a ball bearing, n = the number of rollers or balls, as the case may be and c = a constant = 5, for conditions recited below.

It will be noted that the entire number of balls or rollers is not considered as carrying load, but the total number divided by c. It is clear that not more than half of the balls (or rollers) can be in the loaded side of the journal. It is also clear that that ball immediately in line with the direction of the load will carry most, while the balls (or rollers) at either side will carry consecutively less and less. For bearings that would take between 15 and 30 balls or rollers it will do to take $c = 5$. For other conditions the proportionate load imposed on each ball (or roller) must be figured; this is readily done mathematically or graphically. In many forms of ball and roller bearings the balls (or rollers) are separated, thus their number is less than the possible maximum that might be inserted did they lie in actual contact; n in the above formula is to be taken always as this possible maximum.

Professor Stribeck gives values of p for roller bearings and continuous service from 85 to 155 lb.

Fig. 69 is taken from Stribeck's experiments and shows the relationship

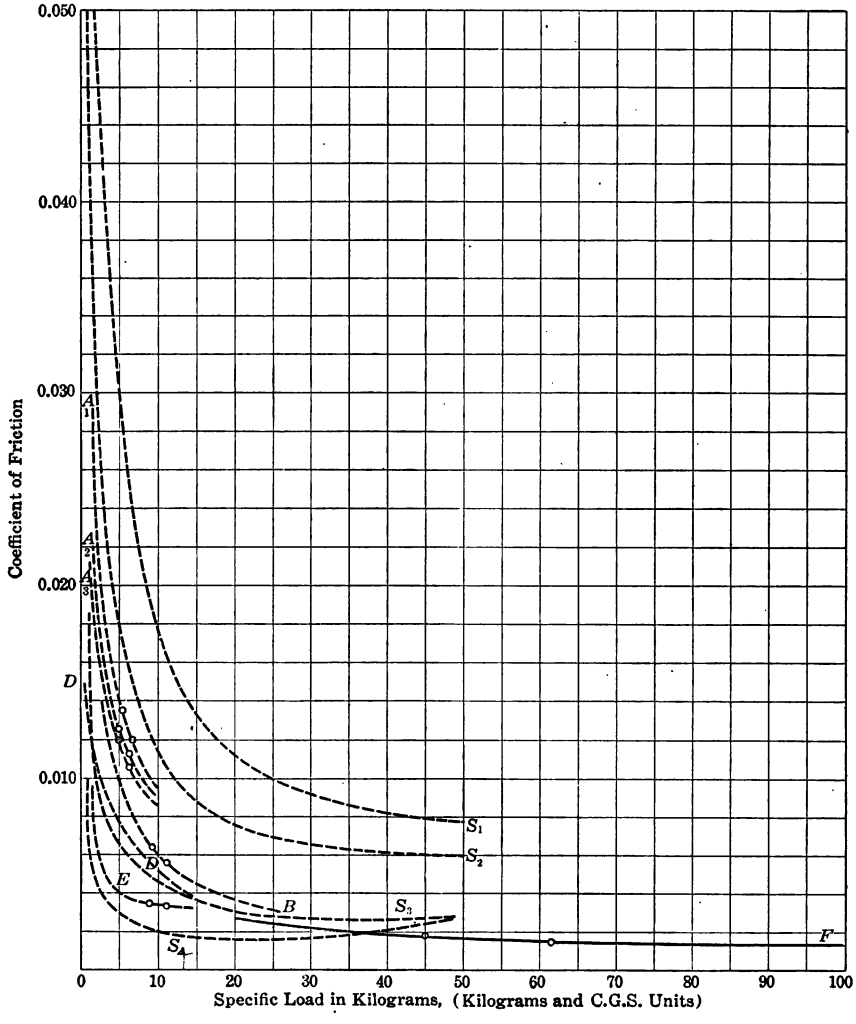


FIG .69. RELATIONS OF COEFFICIENTS OF FRICTION FOR PLAIN, ROLLER AND BALL BEARINGS (*Stribeck*).

between the coefficient of friction and unit load for journal bearings, several types of roller bearings and one type of ball bearing. Referring to the specific curves: Curves S are for a babbitted journal—S_1 at 1100 revolutions per

minute, S_2 at 380, S_3 at 64 and S_4 at 12. Curves A are for rollers in a bronze cage, circumferentially fitting and supporting each roller throughout its length. A sleeve on the shaft and a liner in the box carry the rolls. Hard and soft rollers were tested; A, at 1100 revolutions per minute, A_2 at 285, A_3 at 190. Curves B are for hollow rollers on loosely fitted pins that connect the end cages. No liners are used on the shaft nor in the box. The speeds are 190 to 760 revolutions per minute. Curves D are for hollow, flexible, short, sheet steel rollers on loosely fitted pins that connect the end cages. No liners were used on the shaft or in the box. The speed was 1100 revolutions per minute. Curves E are for flexible spiral rollers, placed loosely in the box and alined by three longitudinal bars connecting the end cages. No liner for the shaft or box are employed. The speeds are 56 to 1100 revolutions per minute. Curve F is for a ball bearing of the two-point, curved ball-race type (with race curvature but slightly in excess of ball curvature) at 65 to 1150 revolutions per minute. The range of permissible specific load, as indicated by the appearance of the bearing after long continued runs, is shown by small circles on the curves.

The above curves and description are taken from a translation and amplification of the work of Professor Stribeck, made by Henry Hess, and found in the *Trans. A. S. M. E.*, Vol. XXVII. That celebrated work lays the foundation for the intelligent design of ball and roller bearings.

As ball and roller bearings are made by only a few firms and as many of the devices are patented it will be necessary to refer frequently to specific bearings in the following pages.

PROFESSOR STRIBECK'S INVESTIGATION

Ball bearings first came into general use in the bicycle. They were of the so-called cone-bearing type and in general had an adjustable cup and cone and two-point contact. They were designed by rule of thumb and the basic principles were in obscurity.

This was the condition of affairs in 1898, when the problem of determining the carrying capacity of balls and the laws for the design of ball bearings was attacked by Professor Stribeck, then director of the Scientific Technical Experimental Laboratories at Neu-Bablesberg, Germany. His solution set a thorough precedent.

He first determined the load sustaining capacity of balls and found that it is proportional to the square of the ball diameter, and very much dependent upon the shape of the surfaces or grooves between which the ball is held.

CURVATURE OF THE RACE GROOVES

The governing principle in regard to the shape of the races is that the ball shall roll over them and not slide. Theoretically balls touch their contact sur-

faces in points only. Practically, even with balls of hardened steel of high elasticity they touch over surfaces of appreciable area instead of at points. Theory and experiments both prove that the two-point contact is the best. To-day all of the better known ball bearings and most of the ball-thrusts have two-point contact.

If it were possible to so shape the groove that all parts would sustain load equally we would have a condition of maximum sustaining capacity of the ball. Such a groove would be represented by a profile semi-circular in section with radius of curvature equal to one-half the ball diameter. Because of excessive friction this form is not practicable. It is evidently one limit of the curvature of the race.

Turning to the other limit, if we begin with a cylindrical race the sustaining power increases without a proportinate increase of friction as the curvature is increased up to the point of equality of curvature, when sliding sets in and with that a change from rolling to sliding friction. The best results come from proportions in which $r_1 = 9/16\ D$ and $r_2 = 25/48\ D$. Here r_1 = the radius of curvature of the outer race groove, r_2 = the radius of curvature of the inner race groove, and D = the ball diameter. The radius for the outer groove will be seen to be slightly greater than the other, the reason for this being that the concavity of the outer race tends to embrace more of the ball than the inner race, which is convex. Even greater approximation to equality of race and ball curvature is desirable, but is impracticable owing to workshop difficulties.

LOAD CARRYING CAPACITY OF RADIAL BEARINGS

Early investigators of ball bearings took into consideration the crushing strength of the balls. This is an erroneous starting-point. The capacity of a ball in a ball bearing is not its crushing strength by any means, but it is its resistance to permanent deformation; some deformations take place with any appreciable load no matter how high grade the material from which the balls are made. The load carrying capacity of a ball bearing is directly proportional to the number of balls and to the square of the ball diameter. According to Stribeck this capacity P is expressed by the equation $P = k\ n\ D^2$ in which n = the number of balls, D the ball diameter, and k a cofficient depending upon the type of bearing, the material from which it is made and the speed at which it is run.

Mr. Hess gives values of this factor k for a special steel used by his firm as follows, provided that D is expressed in units of an eighth of an inch:

For uninterrupted race grooves with cross-sectional curvature as mentioned above, separated balls, uniformly distributed load and a uniform speed up to 3,000 revolutions per minute, $k = 9$.

For full type bearings with the filling opening in one race at the unloaded side, otherwise as above, $k = 5$.

For both ball tracks interrupted by filling openings, inelastic cage separators for the balls, or for full ball type; and speeds not over 2,000 revolutions per minute with a uniform distributed load, $k = 2.5$.

For thrust load on a radial bearing of the first type in this tabulation $k = 0.9$.

In general, the larger the number of balls, the smaller the value of k. The radial load bearing is up to high speeds, practically unaffected by the speed as to its carrying capacity.

LOAD CARRYING CAPACITY OF THRUST BEARINGS

Ball thrust bearings are made in three general types. In the first both races are flat, in the second one race is flat the other grooved. In the third both races are grooved. The load carrying equation given by Mr. Hess is:

$$P = \frac{k_1 n D^2}{\sqrt[3]{S}}$$

In which $P =$ the load capacity in pounds, $k_1 =$ a factor determined by experiment and varying with the material and the shape of the ball track, $n =$ the number of balls, $D =$ the ball diameter and $S =$ the number of revolutions per minute of the bearing. Up to about 3000 revolutions per minute the effect of speed varies with its cube root.

Mr. Hess gives the following values for k_1.

For material such as used by his firm and for races having grooves with a cross-sectional radius equal approximately to 0.82 D, $k_1 = 25$ to 40.

For unhardened steel, such as is occasionally used for very large races and where there is no hammering or no sharp blows, $k_1 = 0.5$.

When one or both races are flat k_1 should be reduced to one-fourth the above value.

The Standard Roller Bearing Company gives the following load capacity formulas for ball thrust bearings having a groove in each washer, stating that the ratings obtained from their use are very conservative and give a condition of loading under which a bearing can be guaranteed if properly installed and lubricated.

The formula for the light type bearing with 17 balls is:

$$P = 32{,}000 \frac{n D^3}{a\sqrt{S}}$$

in which $P =$ the load capacity in pounds,

$D =$ the ball diameter in inches,

$n =$ the number of balls,

$a =$ the pitch diameter of the ball grooves in inches,

$S =$ the speed of the shaft in revolutions per minute.

The formula for the medium and heavy bearings having 11 and 9 balls respectively is:

$$P = 19{,}200\frac{n D^3}{a\sqrt{S}}$$

The notation is the same as given above.

Ball thrust bearings are commonly of the two-point type to which all of the preceding formulas apply. However, a line of thrust bearings having four points of contact is made by the Auburn Ball Bearing Company.

SECTION II

CONSTRUCTION OF BALL BEARINGS

From the fact that the load carrying capacity of a ball bearing is directly proportional to the number of balls, early designs were of the full type with an opening to admit the balls. Fig. 70 shows an entering notch at the side cutting through into the groove in the upper race. This notch was closed by means of a block, as shown at the left in Fig. 71. A modification of the same idea was

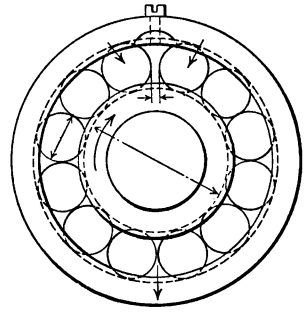


FIG. 70. RADIAL BALL BEARING WITH BALL ENTERING NOTCH IN UPPER RACE.

a hole through the outer race closed by a screw plug as shown at the right in Fig. 71. In both of these constructions the closing piece was located at the point of least pressure. A modification of the first device used in small bearings consisted in stopping up the notch at the side by means of a small screw. These early devices are credited to Riebe, Bierschenk and Bruhl.

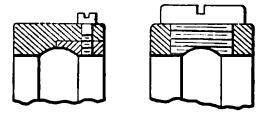


FIG. 71. DEVICES FOR CLOSING THE ENTERING NOTCH IN BALL-BEARING RACES.

All of these constructions reduce the load carrying capacity of the bearing because of the weakened race section where the notch or opening is made and because after slight wear there develops an interference between the balls and the edges of the opening. Furthermore this type of bearing cannot take any thrust load, and in practically all constructions radial bearings are required to

take a limited amount of thrust because of the end play of the shafts that they support.

To overcome these obvious defects the partly filled type was developed. Here the balls are separated by means of spacing blocks or cages. As it is desirable to enter as many balls as possible two general constructions have grown up. The first is known as the eccentrically filled bearing, in which the inner race is brought in contact with the outer race, leaving a crescent shaped opening between them into which all of the balls but one are entered. They are then rolled around until the outer and inner races are nearly concentric and the last ball is snapped into place under a slight pressure. The races are perfectly continuous and there is nothing structurally to interfere with the free, perfect rolling of the balls. This is the D. W. F. and Hess-Bright Construction.

Conrad also patented a process in which the outer ring is heated to expand it, or the inner cooled to contract it, or both operations performed simultaneously. With this process it is not possible to introduce more than two or at the most three more balls than if no such device is resorted to.

The other type is made with two modifications. In the first there are two filling notches, one in the outer and one in the inner race, sometimes introduced at an angle with the circumference, in which case they are inclined in opposite directions. The balls are filled through these notches. The other modification consists in grinding, on one side, the inner face of the outer ring eccentric with its circumference, so that this ground surface is tangent, or nearly tangent, to the bottom of the groove. This forms an opening through which the balls can be entered. The first construction is used by Fichtel and Sachs, the second by the Rhinelander Machine Works.

THE BALL CAGE

With the use of the partly filled type it became necessary to hold the balls in positions equally spaced around the race. In fact, it is practically impossible so to proportion races and balls, even of the full-ball type, that the latter exactly fit the circumference. Furthermore, the cage has two other important uses. The one to reduce noise, the second to prevent wear.

If a full ball type of bearing is set in motion under the usual vertical loading, whenever a ball passes from the region of load to the region where it is free from pressure it is snapped forward with a motion analogous to the snapping of an apple seed from between a boy's fingers. It strikes forcibly against the adjacent ball and inner race, bounds back against the outer race and continues to vibrate. Before it comes to rest it is struck by the following ball which is then being subjected to the same action. This is a chief cause of noise in ball bearings.

The second trouble that the cage overcomes is wear of one ball on another. With full type bearings, it is not uncommon to find balls each of which has a groove worn around its circumference with a radius of curvature equal to that of the ball diameter. One cause of this is the deflection of the shaft or journal, as in the case of a large gear which bends its shaft when its normal load or an overload is suddenly applied. The inner race of the bearing is turned obliquely with reference to the outer race.

If the axis of the outer and inner races are co-incident, the path of the ball is a true circle. If, however, we twist one race into an oblique position with reference to the other and plot the ball path, we will find that it is no longer a circle, but a compound curve with two cusps. Thus in one revolution of such a bearing, the balls approach and recede from each other twice. This action interferes with the rolling action and causes sliding and pressure between balls and balls in full bearings; and between balls and cages where the latter are used. As the cages are stationary with reference to the balls and in the best forms also yield slightly this "binding" tendency is practically eliminated.

It is important to keep all ball bearings properly alined.

Cages were devised to keep the balls properly separated in a partly filled type bearing, to reduce noise and to prevent certain forms of ball wear. One of the most perfect theoretical forms is that of Conrad which consisted in separating the balls by a short helical spring with suitable bearing plates. This form was generally used for several years by the Hess-Bright and D. W. F. Companies and is still used as a special form for certain conditions.

Other satisfactory forms consist of a cage made of brass or some form of bearing metal and so constructed as to be somewhat elastic. Many ball bearing failures can be traced to improperly designed and constructed cages, in fact it is not unusual in a failed bearing to find a cage literally ground to pieces. It is not a difficult matter to construct a cage which shall be satisfactory for a given set of conditions, but ball bearings are nowadays manufactured for stock in many thousands per day; it is therefore impossible to know whether a given bearing will be used at a few or many thousands of revolutions per minute, under uniform or variable loads or speeds, under no or great shock, etc. No cage has yet been found that will respond with equal satisfaction to all of these many conditions.

Many attempts have been made to introduce a rolling member between the balls of ball bearings and the rollers of roller bearings. These are used instead of cages and are based on the conception that there will thus be only rolling contact between the various rolling elements and that pressure between them will then be rendered harmless. As a matter of fact there can be no pressure contact between balls or balls and separators in a correctly acting bearing. Such pressure is due either to faulty original design or to faulty action

from deflections, etc. In either case spinning or sliding will be caused and rolling prevented. It is clear that an element that must roll to fulfil its mission cannot do so when not allowed to roll at the critical periods.

Both balls and rollers have been used for this purpose.

FRICTION OF RADIAL BALL BEARINGS

A great deal has been written in regard to the smallness of the coefficient of friction in ball and roller bearings. Undoubtedly this feature has been given much more importance than it deserves. No argument or reiteration is necessary to prove that bearings with rolling contact have less friction than bearings with sliding contact.

It is impossible to generalize in regard to such coefficients of friction.

In a radial bearing the nature of the loading has a direct influence upon the shape of the friction curve. Figure 72 reproduced from the *American Machinist*

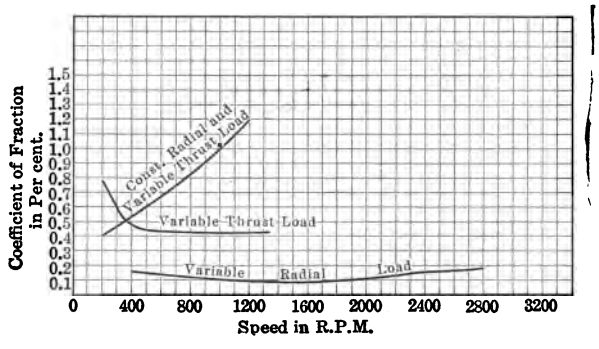


FIG. 72. TYPICAL CURVES FOR A NO. 308 RADIAL BALL BEARING UNDER VARIOUS CONDITIONS OF LOADING.

of March 4, 1909, shows three curves. The first is for a variable radial load, the second for a variable thrust load, the third for a combination of constant radial load and variable thrust load all on a radial ball bearing. Tables 40, 41, and 42 show the logs of the tests from which the curves were plotted. These tests were made on a special ball bearing testing machine, designed by Mr. Hess and built by Riehle Brothers Testing Machine Company. These coefficients and curves are characteristic of this type of bearing.

Under a constant radial load the coefficient of friction for a given speed is nearly constant. With a continued constant radial load and variable thrust load, the coefficient increases in value and the curve rises rapidly. This condition is caused by a sliding action which increases in degree with an increase in the thrust load. With a variable thrust load only, the curve starts high, drops sharply and then continues at nearly a constant value.

It must not be argued from this one set of curves that radial bearings should not be used under thrust. For speeds roughly below 1500 revolutions per minute the thrust type of bearing is best; at about this speed (the exact point varies with changes in size) the curve of the radial type under thrust crosses and falls below that of the pure thrust type.

It will thus be good engineering practice to use a single radial bearing to carry both radial and thrust load, or to use one radial bearing to carry radial load and a second one to carry thrust load only, which it is made to do by freeing its outer race circumferentially and allowing it to rest against its radial jaws only, or to use instead of this second radial bearing a separate bearing of the pure thrust type.

Radial load lb.	Vernier	Vernier	Difference	Average	Coefficient of friction in per cent.	Oil temperature, degrees Fahrenheit
400	44875	44661	0.00214	0.00107	0.1633	
600	44851	44652	0.00199	0.000995	0.1517	
800	44663	44489	0.00174	0.000870	0.1325	79.1
1000	44581	44390	0.00171	0.000855	0.1302	
1200	44602	44439	0.00163	0.000815	0.1240	
1400	44524	44366	0.00158	0.00079	0.1200	
1600	44506	44349	0.00157	0.000785	0.1195	82.5
1800	44443	44276	0.00167	0.000835	0.1272	
2000	44467	44281	0.00186	0.00093	0.1416	84.0
2400	44452	44231	0.00221	0.001105	0.1686	
2800	44469	44229	0.00240	0.00120	0.1830	85.5

Room temperature, 76° F.
Speed, 325 r. p. m.
Bearing, No. 308. Bore 1.5748 in.
Rated radial load, 1450 lb.

TABLE 40.—LOG OF TEST ON A NO. 308 RADIAL BEARING UNDER A VARIABLE RADIAL LOAD

Variable thrust load lb.	Constant radial load lb.	Vernier	Vernier	Difference	Average	Coefficient of friction in per cent.	Oil temperature in degrees Fahrenheit
200	400	43846	43318	0.00528	0.00264	0.402	72.10
400	400	43932	43222	0.00710	0.00355	0.541	
600	400	44011	43109	0.00902	0.00451	0.687	75.7
800	400	44117	43018	0.01099	0.005495	0.837	
1000	400	44259	42881	0.01378	0.00689	1.050	79.30
1200	400	44326	42763	0.01563	0.007815	1.191	
1400	400	44443	42596	0.01847	0.009235	1.406	84.5

Room temperature, 73°. F.
Speed 325 r. p. m.
Bearing, No. 308.

TABLE 41.—LOG OF TEST OF A NO. 308 RADIAL BEARING UNDER A CONSTANT RADIAL AND VARIABLE THRUST LOAD.

Thrust load lb.	Balance Weight	Balance Weight	Average	Torque	Coefficient of friction in per cent.	Oil temperature degrees Fahrenheit
200	0.132	0.125	0.128	1.262	0.803	73
400	0.158	0.148	0.153	1.509	0.479	
600	0.218	0.201	0.209	2.061	0.436	76
800	0.288	0.267	0.277	2.732	0.434	
1000	0.355	0.343	0.349	3.440	0.437	80
1200	0.412	0.413	0.412	4.061	0.431	
1350	0.446	0.485	0.466	4.600	0.433	85

Room temperature, 74° F.
Speed, 325 p. m.
Bearing, No. 308.

TABLE 42.—LOG OF TEST OF A NO. 308 RADIAL BEARING UNDER A VARIABLE THRUST LOAD.

Certain forms have also been devised carrying two rows of balls, so arranged that original radial load develops thrust components and *vice versa*. Such a bearing is thus capable of carrying also simultaneous thrust and radial loads. While ball bearings all demand a very high degree of accuracy, this need is greatly accentuated for such two row construction. A location which demands a bearing of this kind is in the hubs of automobile wheels.

A word should be said about the mounting of radial bearings. Good prac-

tice calls for a fit of the inner race upon the shaft, such that it will not slide but rotate with it as a single member. Keying is objectionable because of the weakening of the race cross-section. Approved practice consists in making the inner race a pressing fit on the shaft and binding it laterally by means of jam nuts or some equivalent device. A hammer should never be directly employed in contact with a ball bearing. Nor should the inner race ever be

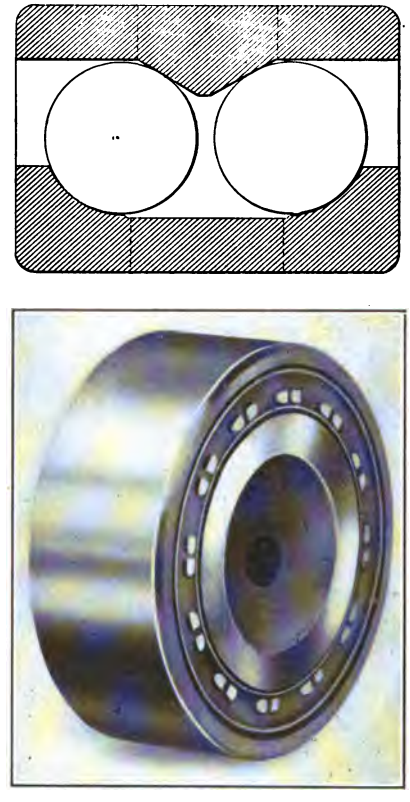


FIG. 73. HESS-BRIGHT DOUBLE-ROW BALL BEARING.

forced home by pressure, or blows applied to the outer one, acting through the balls. If the outer race slides within its housing no harm is done. For that reason such a fit is usually made a light pressing fit or a "sucking fit" with a little freedom sidewise. In fact such sliding is beneficial and is generally insisted in by ball bearing makers as the slow creep gradually brings all portions of this outer race around under the load and so distributes its effect over the whole of the outer race track.

TWO ROW RADIAL BALL BEARINGS

For reasons cited on page 185 several developments of double row or two-row bearings have appeared. Fig. 73 shows such a bearing by Hess-Bright. This consists of an inner race, an outer race, two rows of balls and a ball separator. The inner ball track is concaved, the outer one flat, but inclined at an angle of 30° to the axis. The inner race radius center lies in the line passing through the ball center normally to the outer race face. The flat inner race allows an even distribution of load between the two rows.

Fig. 74 shows a bearing of this kind made by the New Departure Manufacturing Company. The balls are staggered, fit into a two-piece outer race, one-piece inner race, and the parts are assembled by a soft outer steel shell.

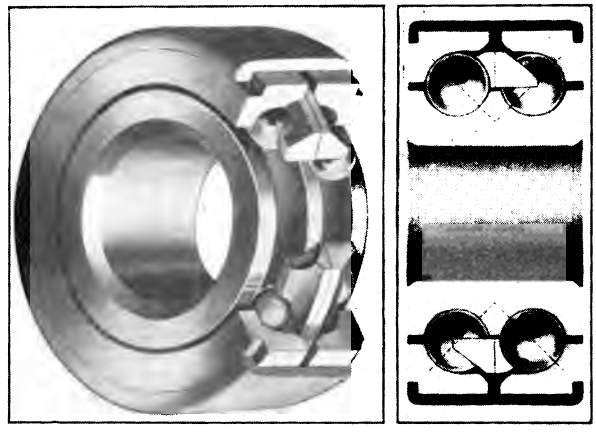


FIG. 74. TWO-ROW BALL BEARING OF NEW DEPARTURE MANUFACTURING COMPANY.

Fig. 75 shows a Swedish bearing of the S. K. F. Ball Bearing Company. The novel feature is the spherical bearing surface of the outer race which makes the bearing self alining, for the center of this spherical surface lies on the axis of the bearing. The method of caging the balls is shown as well as the method of assembling.

It must not be assumed that carrying capacity is doubled by doubling the number of balls. Workmanship of the necessary accuracy is not feasible. When accuracy of workmanship is depended upon solely for load distribution, a liberal factor must be employed in reduction; when in addition races of small curvature as compared with the ball grooves are used this factor must be still further liberalized. See Section I, page 176 relating to curvature of race grooves.

CONE-TYPE BALL BEARINGS

Another type that is used in small sizes, as for magnetos, is a modification of the cone-type bearing of bicycle days. It is particularly serviceable where assembling is difficult, for the inner race can be slipped into its seat, the ball filled cage then inserted and the outer race added.

This type must be adjusted with care, but it is a fallacy to think that adjustment can compensate for wear.

CAGES FOR THRUST BEARINGS

Experiment has shown that the balls in thrust bearings will run against each other in a manner similar to radial bearings if they are free, thus it is customary to cage the balls. Mechanically such a cage has another use in that it assembles the balls into a unit, permitting them to be easily handled in the operations of shipping and inserting in the bearing housing. In order that each ball will take its proportionate share of the load, it is essential that at least one of the races should be provided with a spherical groove.

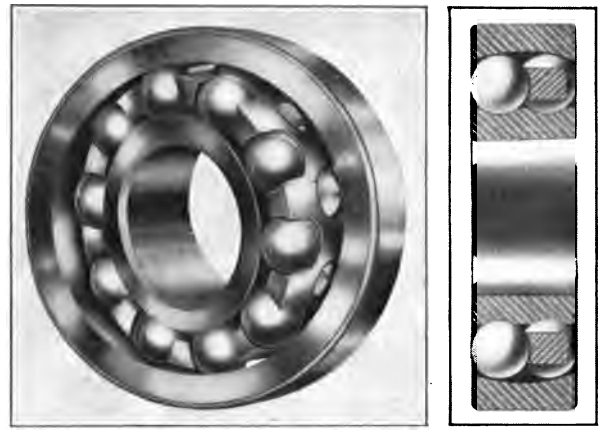


FIG. 75. S. K. F. TWO-ROW RADIAL BALL BEARING.

That the bearing may adjust itself to its load, it is common practice to make the lower race or ring with a spherical surface entering a spherical seat in the machine housing, or in an adaptor ring.

The advantage of the latter construction lies in the difficulty with which a spherical seat is machined in large castings.

Experiments have shown that the theoretical advantages of such a construction are realized in practice.

FRICTION OF THRUST BALL BEARINGS

Fig. 76 shows a typical curve for a thrust ball bearing under a variable thrust load.

Figs. 77 and 78 show a two-row thrust bearing made by the S. K. F. Ball Bearing Company. Because of the spherical casing it is self alining. It will take thrust in either direction.

ACCURACY OF BALL BEARINGS

There is no other machine element produced in large numbers which calls for such accurate workmanship as do high-grade ball bearings. The accuracy is almost painful. No details can be overlooked or slighted.

At the outset the material must be suitable in both balls and races, the parts must be properly hardened and tempered, and the grinding of the race grooves and balls must be accurate to within 0.0001 in. It is a practical impossibility

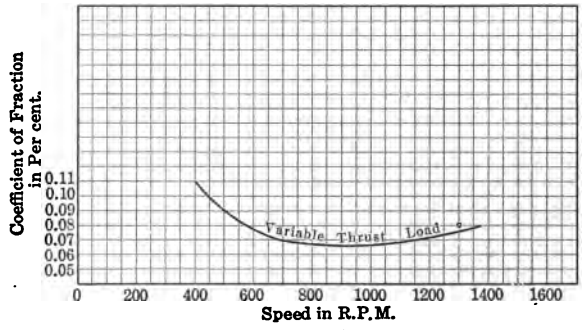


FIG. 76. TYPICAL CURVES FOR A SMALL THRUST BEARING UNDER VARIABLE LOAD.

commercially to grind all balls to a limit of 0.0001 in. In practice, therefore, they are sorted into grades and the allowable limit of variation in each grade is 0.0001 in. When a bearing is assembled, care is taken not to mix balls of different grades. Only by adopting this course is it possible to produce a bearing in which the load will be properly distributed between the balls with an assurance that one ball if large may not at times take all of the load, or, if small, receive none, allowing its neighbors to carry its share. Furthermore, it is the only way in which to produce a ball bearing that will run quietly. Irregularity among the balls themselves is a fruitful source of noise; possibly second only to inaccuracies in the races. As a general rule, however, the greater the inaccuracies in both races and balls, the greater will be the noise in running. Only by having balls of practically equal diameter can a bearing be made that will run quietly.

This indicates the degree of accuracy required of the bearing manufacturer. There is another side to the question, however, and that is the accuracy of the parts of the machine in which the bearing is fitted.

If the two-part housing in which the outer race is held does not fit, it is more than likely that if tightly clamped, it distorts the race and presses upon the



FIG. 77. S. K. F. DOUBLE THRUST BALL BEARING.

balls, to the detriment of the running of the bearing and perhaps causing its failure. Again, if the shaft is too much larger in diameter than the bore of the inner race, the latter may be expanded slightly and the balls pinched. These are two common causes of failures in ball bearings.

Another trouble is overloading. This is the chief cause of abrasions. If the abrasions are in the middle of the groove of either a radial or thrust bearing,

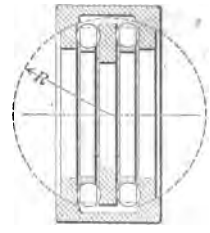


FIG. 78. SECTION OF S. K. F. DOUBLE THRUST BALL BEARING.

they probably indicate that the load in a normal direction is too great. If abrasions are on one side of the groove in a radial bearing, they indicate that the thrust load is too great. If the track of the balls in a radial bearing swerves from side to side, it indicates deflection of shaft or bearing, or original misalinement.

MATERIALS FOR BALL BEARINGS

The material used for ball bearings for machinery is steel. In the higher duty bearings, both balls and races are made from special alloy (tool) steel that has a very high elastic limit after hardening and drawing.

In some bearings, both balls and races are made of a low carbon steel with case-hardened surfaces. For severe service such bearings have not proved successful. The elastic limit of the material is much lower than of the alloy (tool) steel, and in the ball there is a tendency for the hard shell to separate from the softer core; this is in evidence also in the races.

In very large thrust ball bearings a high carbon unannealed tool steel is sometimes used without hardening. This is because of the difficulty and expense of hardening and grinding such large rings without the accompanying losses from warping and checking and the difficulty of getting perfect uniformity of surface hardness. Hess-Bright (D. W. F.), however, produce races up to 54 in. diameter of a special alloy steel that are hardened throughout, as are their smaller ones.

PROPORTIONS OF BALL BEARINGS

The proportions of ball bearings are well standardized. Oddly enough, ball diameters are universally expressed in inches and fractions thereof, while the dimensions of the races are given in either millimeters or English units. German builders use millimeters with a single exception where a firm has developed a series in English units, adapting them for the British trade, though that also uses chiefly ball bearings in millimeters. Even in England most ball bearings are made to millimeters as is also the more general practice of American manufacturers who have followed the German example. This general adoption of the millimeter dimensions is due to the fact that early German makers rehabilitated the ball bearing by the development of the principles and construction data of the modern type and secured a wide vogue for their products that made the sizes standard. As a rule each manufacturer makes a wide and narrow type of radial bearing, with three series for each; namely, light, medium, and heavy. In some cases a fourth series has been standardized, known as "extra heavy."

Thrust bearings are usually made in three series, light, medium, and heavy. The heavier the bearing the larger the balls for a given strength.

At the 1911 Spring Meeting of the Society of Automobile Engineers, the standards committee rendered a report containing Tables 43, 44, and 45, giving recommended standard dimensions for the light, medium, and heavy series, respectively, for radial ball bearings. These standards are referred to as "Ball Bearings Standards A." It will be noted that the last column gives a

radial load rating in pounds for each bearing. In this connection the committee reported:

"Attention is called to the fact that the capacities given in the tables are based upon ball bearings manufactured of suitable workmanship and of suitable material and running at uniform speed and uniform radial load, the speed not exceeding 500 revolutions per minute."

"It is further suggested in explanation of the load standards that it cannot be expected that all conditions will be covered by the loads given. For conditions of shock, actual thrust, and a combination of the two, greater factors of safety will have to be used."

Table 46 is taken from this same report and shows the tolerances or limits worked to by many bearing manufacturers as embodied in statements made to the committee. The table applies to outer race diameter, bore and width.

Other standard dimensions are not given here for they can be found in manufacturers' catalogues.

No.	Bore		Diameter		Width		Corner at bore of inner race		Radial load in lbs.
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	
200	10	0.39370	30	1.18110	9	0.35433	1	0.04	120
201	12	0.47244	32	1.25984	10	0.39370	1	0.04	140
202	15	0.59055	35	1.37795	11	0.43307	1	0.04	160
203	17	0.66929	40	1.57481	12	0.47244	1	0.04	250
204	20	0.78740	47	1.85040	14	0.55118	1	0.04	320
205	25	0.98425	52	2.04725	15	0.59055	1	0.04	350
206	30	1.18110	62	2.44095	16	0.62992	1	0.04	550
207	35	1.37795	72	2.83465	17	0.66929	1	0.04	600
208	40	1.57481	80	3.14962	18	0.70866	2	0.08	860
209	45	1.77166	85	3.34647	19	0.74803	2	0.08	950
210	50	1.96851	90	3.54332	20	0.78740	2	0.08	1000
211	55	2.16536	100	3.93702	21	0.82677	2	0.08	1160
212	60	2.36221	110	4.33072	22	0.86614	2	0.08	1550
213	65	2.55906	120	4.72443	23	0.90551	2	0.08	1670
214	70	2.75591	125	4.92128	24	0.94488	2	0.08	1820
215	75	2.95277	130	5.11813	25	0.98425	2	0.08	2130
216	80	3.14962	140	5.51183	26	1.02362	3	0.12	2650
217	85	3.34647	150	5.90554	28	1.10236	3	0.12	2850
218	90	3.54332	160	6.29924	30	1.18110	3	0.12	3400
219	95	3.74017	170	6.69294	32	1.25984	3	0.12	3750
220	100	3.93702	180	7.08664	34	1.33858	3	0.12	3950
221	105	4.13387	190	7.48035	36	1.41732	3	0.12	4600
222	110	4.33072	200	7.87405	38	1.49607	3	0.12	5000

TABLE 43.—PROPORTIONS OF RADIAL BALL BEARINGS—LIGHT SERIES.

No. of bearing	Bore		Diameter		Width		Corner at bore of inner race		Radial load in lbs.
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	
300	10	0.39370	35	1.37795	11	0.43307	1	0.04	200
301	12	0.47244	37	1.45669	12	0.47244	1	0.04	240
302	15	0.59055	42	1.65355	13	0.51181	1	0.04	280
303	17	0.66929	47	1.85040	14	0.55118	1	0.04	370
304	20	0.78740	52	2.04725	15	0.59055	1	0.04	440
305	25	0.98425	62	2.44095	17	0.66929	1	0.04	620
306	30	1.18110	72	2.83465	19	0.74803	2	0.08	860
307	35	1.37795	80	3.14962	21	0.82677	2	0.08	1100
308	40	1.57481	90	3.54332	23	0.90551	2	0.08	1450
309	45	1.77166	100	3.93702	25	0.98425	2	0.08	1750
310	50	1.96851	110	4.33072	27	1.06299	2	0.08	2100
311	55	2.16536	120	4.72443	29	1.14173	2	0.08	2400
312	60	2.36221	130	5.11813	31	1.22047	2	0.08	2800
313	65	2.55906	140	5.51183	33	1.29921	3	0.12	3300
314	70	2.75591	150	5.90554	35	1.37795	3	0.12	4000
315	75	2.95277	160	6.29924	37	1.45669	3	0.12	4400
316	80	3.14962	170	6.69294	39	1.53544	3	0.12	5000
317	85	3.34647	180	7.08664	41	1.61418	3	0.12	5700
318	90	3.54332	190	7.48035	43	1.69292	3	0.12	6400
319	95	3.74017	200	7.87405	45	1.77166	3	0.12	7000
320	100	3.93702	215	8.46460	47	1.85040	3	0.12	7700
321	105	4.13387	225	8.85830	49	1.92914	3	0.12	8400
322	110	4.33072	240	9.44886	50	1.96851	3	0.12	10000

TABLE 44.—PROPORTIONS OF RADIAL BALL BEARINGS—MEDIUM SERIES

No. of bearing	Bore		Diameter		Width		Corner at bore of inner race		Radial load in lbs.
	Mm.	In.	Mm.	In.	Mm.	In.	Mm.	In.	
403	17	0.66929	62	2.44095	17	0.66929	1	0.04	850
404	20	0.78740	72	2.83465	19	0.74803	2	0.08	1050
405	25	0.98425	80	3.14962	21	0.82677	2	0.08	1320
406	30	1.18110	90	3.54332	23	0.90551	2	0.08	1600
407	35	1.37799	100	3.93702	25	0.98425	2	0.08	1900
408	40	1.57481	110	4.33072	27	1.06299	2	0.08	2200
409	45	1.77166	120	4.72443	29	1.14173	2	0.08	2500
410	50	1.96851	130	5.11813	31	1.22047	2	0.08	3400
411	55	2.16536	140	5.51183	33	1.29921	3	0.12	3900
412	60	2.36221	150	5.90554	35	1.37795	3	0.12	4400
413	65	2.55906	160	6.29924	37	1.45669	3	0.12	4900
414	70	2.75591	180	7.08664	42	1.65355	3	0.12	6200
415	75	2.95277	190	7.48035	45	1.77166	3	0.12	6600
416	80	3.14962	200	7.87405	48	1.88977	3	0.12	7300
417	85	3.34647	210	8.26775	52	2.04725	3	0.12	8580
418	90	3.54332	225	8.85830	54	2.12599	3	0.12	10000
419	95	3.74017	250	9.84256	55	2.16536	3	0.12	11880
420	100	3.93702	265	10.43311	60	2.36221	3	0.12	14000

TABLE 45.—PROPORTIONS OF RADIAL BALL BEARINGS—HEAVY SERIES.

Bearing No.	S. A. E. Jan. 1911			Hess Bright			Timken			Fafnir W. Hasselkus			S. K. F. Co. F. H. Poor			Standard Roller Bearing Co. H. M. Hanks			Recommended by users Warner Gear Co. C. E. Davis			Rhineland S. R. Shepard		
Outer race diam.	Plus limit	Minus limit	Total limit	Plus	Minus	Total	Plus	Minus	Total	Plus	Minus	Total	Plus	Minus	Total	Plus	Minus	Total	Plus	Minus	Total	Plus	Minus	Total
200–204 inc... 300–303	.0005	.0005	.001	0	.0006	.0006	.001	0	.001	.0002	.0002	.0004	0	.0008	.0008	.0001	.0001	.0002	0	.0003	.0003	0	.0004	.0004
205–216 304–313 403–411	.0005	.001	.0015	0	.0008	.0008	.001	0	.001	.0003	.0005	.0008	0	.0008	.0008	.0001	.0001	.0002	0	.0003	.0003	0	.0004	.0004
217–222 314–319 412–416	.0005	.0015	.002	0	.0012	.0012	.001	0	.001	.0005	.0010	.0015	0	.0012	.0012	.0001	.0001	.0002	0	.0003	.0003	0	.0004	.0004
Bore																								
200–204 300–302	.001	0	.001	.0002	.0004	.0006	.001	0	.001	.0003	.0002	.0005	.0002	.0004	.0006	.0001	.0001	.0002	0	.0005	.0005	.0002	.0004	.0006
205–216 304–313 403–411	.001	0	.001	.0002	.0004	.0006	.001	0	.001	.0003	.0003	.0006	.0002	.0004	.0006	.0001	.0001	.0002	0	.0005	.0005	.0002	.0004	.0006
217–222 314–319 412–416	.001	0	.001	.0002	.0004	.0006	.001	0	.001	.0003	.0003	.0006	.0002	.0004	.0006	.0001	.0001	.0002	0	.0005	.0005	.0002	.0004	.0006
Width																								
200–204 300–302	0	.002	.002	0	.002	.002	2	....	.....	0	.002	.002	.....	.....	.....	0	.002	.002	....	.....	.....	.001	.001	.002
205–216 304–313 403–411	0	.004	.004	0	.002	.002	.....	....	.....	0	.004	.004	.....	.....	.....	0	.002	.002	....	.....	.....	.001	.001	.002
217–222 314–319 412–416	0	.006	.006	0	.002	.002	.....	....	.....	0	.006	.006	.....	.....	.....	0	.002	.002	....	.....	.....	.001	.001	.002

TABLE 46.—TOLERANCES IN THOUSANDTHS OF AN INCH ALLOWED IN RADIALS BALL BEARINGS BY VARIOUS MANUFACTURERS.

SECTION III

Typical Designs and Mountings for Ball Bearings

In this section are given a number of typical designs and mountings for ball bearings drawn from the practice of a few manufacturers, both foreign and domestic, and applying to various kinds of machines. Obviously other firms could have furnished similar data had there been space to include them.

The first examples are taken from machine tools and indicate the interest that their builders have felt in adapting friction reducing bearings to their product. Figs. 79 to 82, inclusive, are of grinder spindles, ball bearing mounted. Fig. 79 is from an article by W. M. Byorkman, printed in the *American Machinist* of February 9, 1911, page 242.

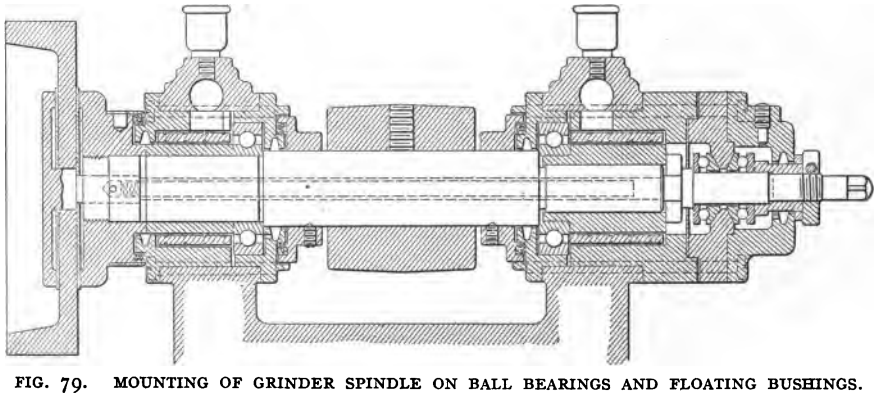


FIG. 79. MOUNTING OF GRINDER SPINDLE ON BALL BEARINGS AND FLOATING BUSHINGS.

It shows a grinder spindle mounted on radial ball bearings with floating bushings. These bushings are used to give steadiness to the spindle as in work where the dimension tolerances are small. They are located close to the ball bearings and have inside and outside clearances of about 0.0002 in. They have end clearances only sufficient to permit of appreciable movement.

The bushings carry no load, but the clearance spaces are filled with oil which acts like a dash pot to suppress the minute vibrations which would otherwise arise in the ball bearings. The final effect is to do away with the "waves" frequently noticed on ground work where the grinding spindle is ball bearing mounted.

The endwise movement of the shaft is prevented by a pair of thrust ball bearings seen at the right hand end.

This particular spindle arrangement was applied to a grinding machine for sizing to thickness type bars of a typewriter.

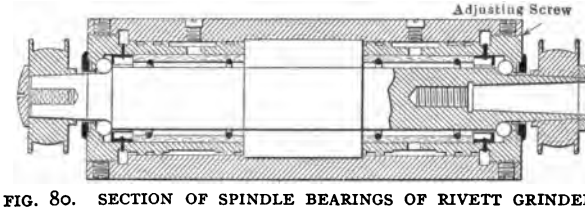


FIG. 80. SECTION OF SPINDLE BEARINGS OF RIVETT GRINDER.

Fig. 80 shows in section the spindle and bearings of a Rivett grinder. From a test made by A. Morin in Paris, France, a speed of 120,480 revolutions per minute has been obtained with a spindle of one of these machines. This is

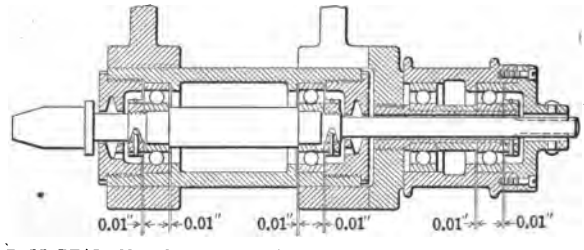


FIG. 81. BALL-BEARING MOUNTINGS FOR HIGH SPEED SPINDLE AND ITS PULLEY.

undoubtedly the highest speed ever recorded for a rotating machine element driven by mechanical power.

Figs. 81 and 82 show two recommended grinder spindle mountings developed

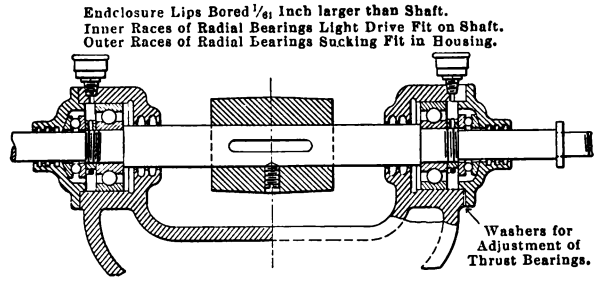


FIG. 82. BALL-BEARING MOUNTINGS FOR BUFFING OR GRINDING HEAD.

by the Hess-Bright Manufacturing Company. In Fig. 81 it will be noticed that the driving pulley is carried on two radial bearings supported by a sleeve independent from the spindle itself, thus the stress of the belt pull does not reach the

spindle. Another feature is in the right hand radial bearing of the spindle itself. The outer race instead of having a groove for the balls has a plain cylindrical surface. Thus the balls have a chance to move in the direction of the length of the spindle if the parts lengthen from heat.

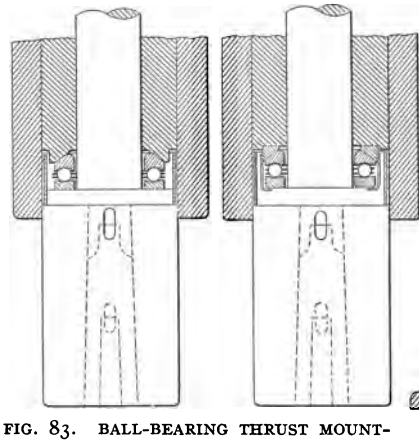


FIG. 83. BALL-BEARING THRUST MOUNTINGS FOR DRILLER SPINDLES.

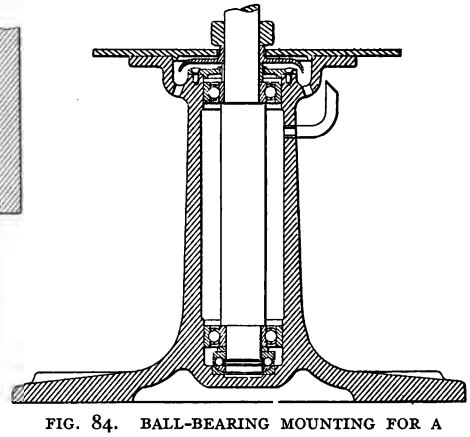


FIG. 84. BALL-BEARING MOUNTING FOR A VERTICAL SPINDLE.

Fig. 82 needs but little comment except to point out the grooves at the ends of the bearing housings provided to prevent the escape of oil and the entrance of dust.

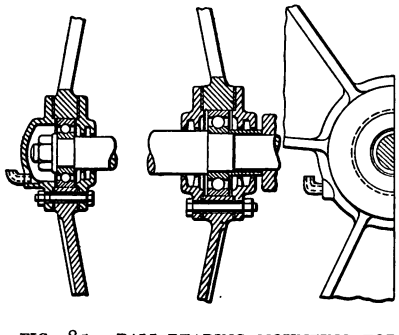


FIG. 85. BALL-BEARING MOUNTING FOR ELECTRIC MOTOR SHAFT.

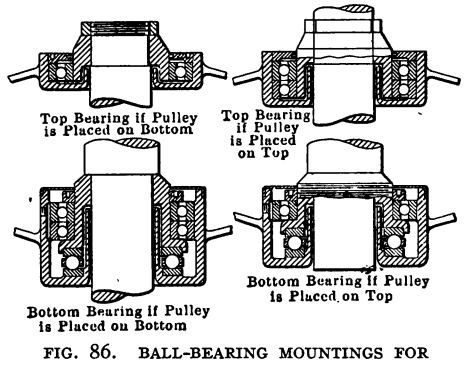


FIG. 86. BALL-BEARING MOUNTINGS FOR VERTICAL ELECTRIC MOTOR.

Fig. 83 shows two mountings for driller spindles and is likewise recommended by the Hess-Bright Manufacturing Company.

Fig. 84 shows a vertical spindle resting on a ball thrust at the bottom and guided by two radial bearings. It is from designs of the Rhineland Machine

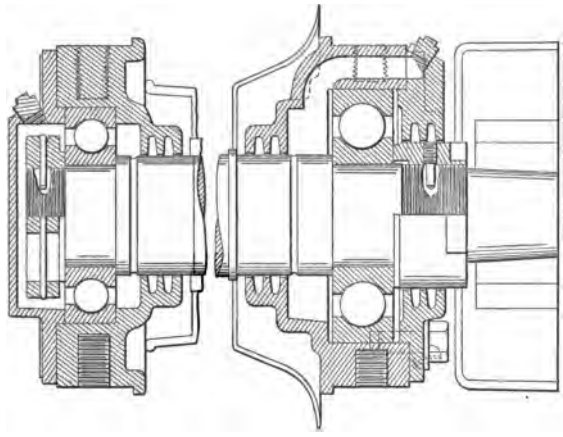


FIG. 87. BALL-BEARING MOUNTING FOR STREET RAILWAY MOTOR SHAFT.

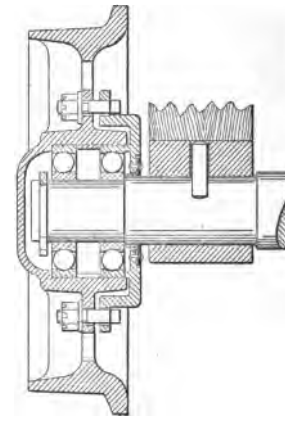


FIG. 88. BALL-BEARING MOUNTINGS FOR TRUCK WHEEL.

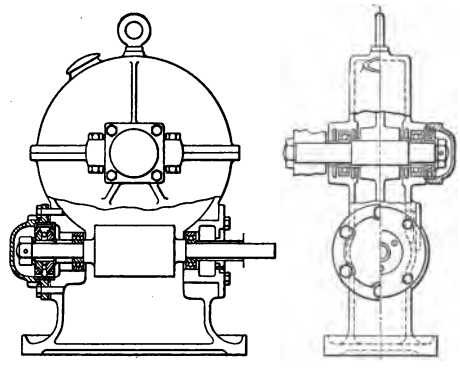


FIG. 89. BALL-BEARING MOUNTING FOR WORM AND WORM WHEEL.

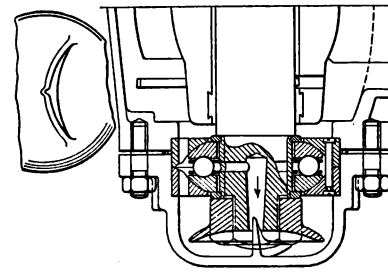


FIG. 90. CENTRIFUGAL OILING DEVICE AND BALL-BEARING MOUNTING FOR A SHAFT STEP.

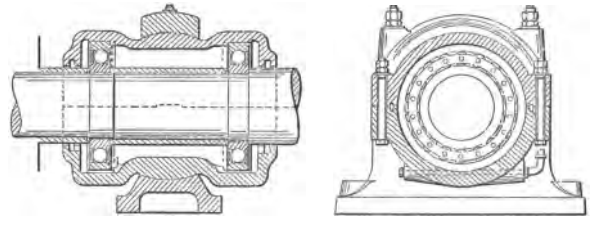


FIG. 91. BALL-BEARING SHAFT MOUNTING WITH SELF-ALINING SEAT.

Works Company. An important point in installing thrust bearings is this: If the upper race is tight on the shaft, the lower leveling washer must be free to a slight extent to allow for movement in preserving alinement. On the other hand, if the upper washer is made with a slight clearance over the shaft diameter the lower ring need not be free.

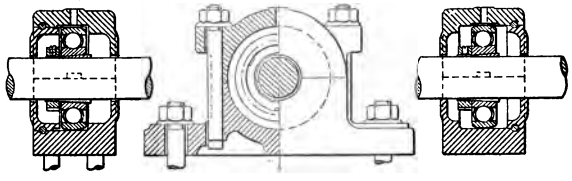


FIG. 92. BALL-BEARING PILLOW BLOCK.

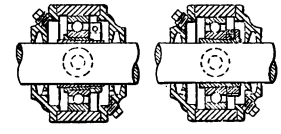


FIG. 93. TWO FORMS OF SHAFT ADAPTERSFOR BALL BEARINGS.

MOTOR BALL BEARINGS

Figs. 85, 86 and 87 show three applications of ball bearings to the shafts of electric motors. Fig. 85 is a Rhineland design for a small sized motor. The clearance in an axially direction of the bearing at the right hand is very plainly

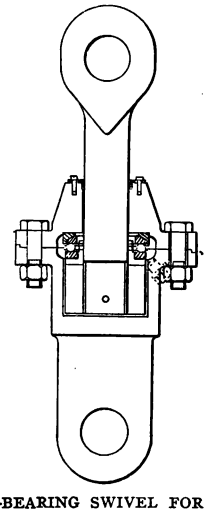


FIG. 94. BALL-BEARING SWIVEL FOR A LIFTING EYE.

shown. Fig. 86 shows two mountings for the shaft of a 45-h. p. vertical motor involving two different combinations of radial end thrust bearings. It is from the practice of the Standard Roller Bearing Company.

Mountings of ball bearings for street railway motors are shown in Fig. 87. A large number of motors with this type of mounting have been in use on some of the street railway lines in New York City. The design is Hess-Bright.

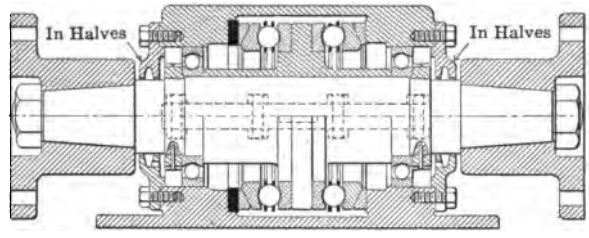


FIG. 95. BALL-BEARING MARINE PROPELLER BLOCK.

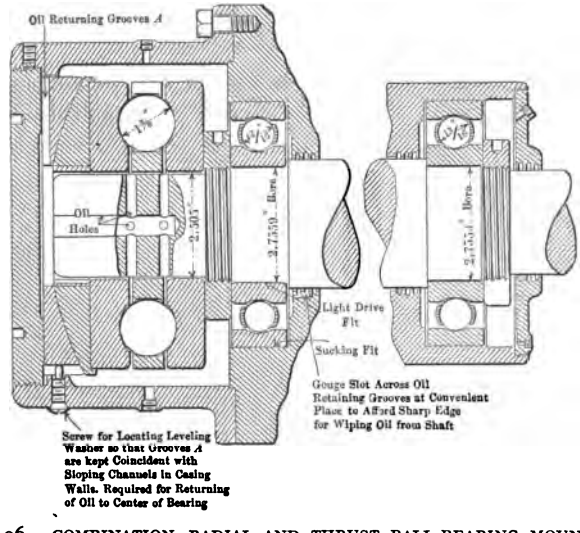


FIG. 96. COMBINATION RADIAL AND THRUST BALL-BEARING MOUNTING.

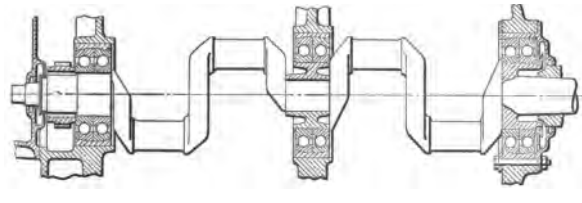


FIG. 97. BALL-BEARING MOUNTING FOR THREE-BEARING, FOUR-CYLINDER AUTOMOBILE CRANK SHAFT.

GENERAL APPLICATIONS

Figs. 88 to 96, inclusive, show a number of ball bearing mountings applied to machinery in general. Of these Figs. 88 to 94, inclusive, are Rhineland designs; Fig. 95 is a Hess-Bright design and Fig. 96 a Standard Roller Bearing design.

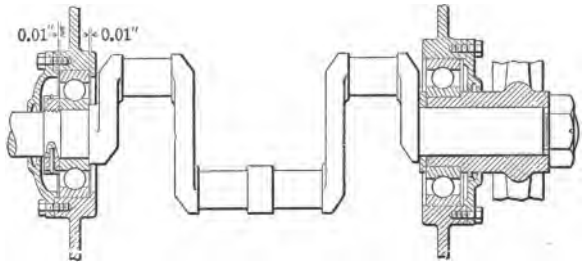


FIG. 98. BALL-BEARING MOUNTING FOR TWO-BEARING, FOUR-CYLINDER AUTOMOBILE CRANK SHAFT.

Fig. 88 shows two radial bearings applied to a heavy truck wheel. Fig. 89 presents the mounting of a worm and worm wheel with both radial and thrust bearings. The thrust bearings are of the Rhineland double thrust type. Fig. 90 is of interest for the design of a centrifugal oiler in a chamber below the step whereby oil is pumped from the lower level of this chamber upward through

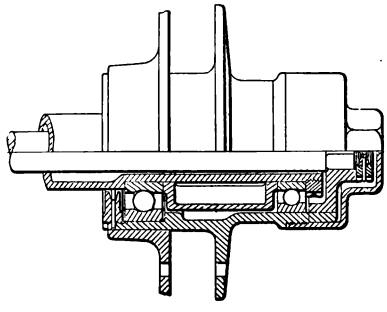


FIG. 99. BALL-BEARING AUTOMOBILE HUB MOUNTING—FLOATING TYPE.

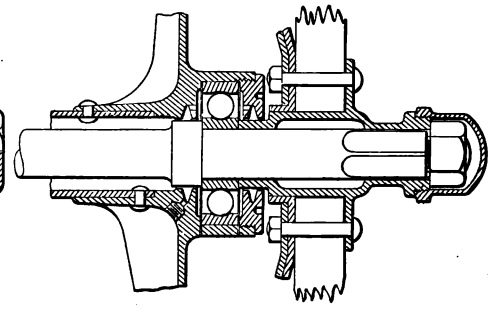


FIG. 100. BALL-BEARING AUTOMOBILE HUB MOUNTING—FIXED TYPE.

the center of the shaft and then outward through channels into the spaces surrounding the balls. The oil finds its way back to the reservoir through an annular space around the bearing.

Figs. 91 and 92 are pedestal box mountings. In Fig. 91 the housing for the bearings has a spherical mid-section fitting a spherical seat in the pedestal frame. This allows for self alinement. Fig. 92 shows the outer races of the

ball bearings provided with spherical surfaces resting against cylindrical surfaces inside of the pedestal frame. It will be noted that the inner surface of the inner ball races is conical and is fitted to the cylindrical shaft by means of an adapter bushing.

Fig. 93 shows two adapter constructions of Hess-Bright design. The one at the right has a combination of two tapered bushings, one surface of each fitting the cylindrical shaft and the cylindrical surface of the inside of the inner race, respectively.

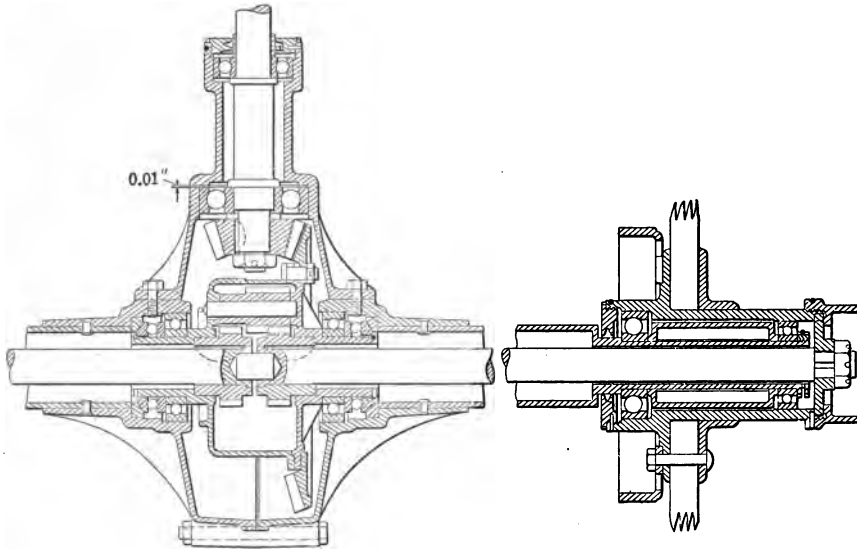


FIG. 101. BALL-BEARING MOUNTING FOR AUTOMOBILE REAR AXLE DIFFERENTIAL AND HUB

Fig. 94 shows a thrust bearing mounted in the swivel of a large lifting eye. The following figure, Fig. 95, illustrates a marine propeller box fitted with two thrust bearings and two radial bearings to carry the load of the shaft. Further description is unnecessary.

The design of Fig. 96 is a large thrust bearing for a 2 1/2 in. shaft combined with two radial bearings to carry vertical load.

A FEW AUTOMOBILE APPLICATIONS

Figs. 97 to 101 inclusive, show a few automobile ball bearing mountings. Of these, Fig. 97 is from the Rhineland Company and the others from Hess-Bright.

Fig. 97 shows three bearings for the crankshaft of a four-cylinder automobile. Each bearing consists of two radial bearings placed side by side and having the

external surface of the outer races of each pair ground to the same radius of curvature. This makes each pair self alining and is a form of two-row ball bearing not mentioned in the previous discussion of this subject.

The other figures in this group do not need particular mention.

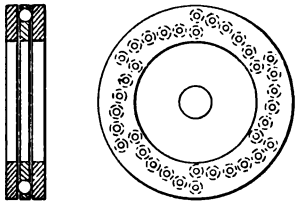


FIG. 102. BALL THRUST BEARING WITH FLAT WASHERS.

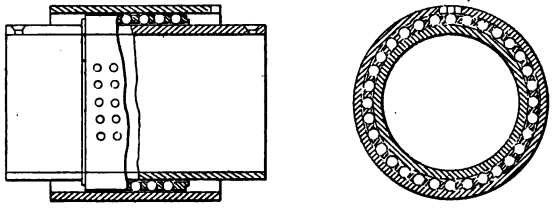


FIG. 103. BALL JOURNAL BEARING FOR FLOATING SHAFT.

The final construction, Figs. 102 and 103, are from the Standard Roller Bearing Company and show, respectively, a plain ball thrust having flat washers—a type previously described—and a special ball journal bearing developed for use on a spindle which had considerable endwise travel.

SECTION IV

Lubrication of Ball Bearings

No better discussion of the lubrication of ball bearings can be found than is given in an article in the *American Machinist*, Vol. XXXIV, page 87, by W. L. Batt. It is given herewith.

Oil is essential for the operation of plain bearings, but it is not so well known that ball bearings likewise require a certain amount of lubrication; it is all too frequently assumed that the latter may run entirely dry.

Contrary to this assumption lubrication has a very positive action in a ball bearing. This action is many fold, existing both between the ball and the path upon which it rolls and between the ball and the separator. Though the principle of the ball bearing is a rolling one, there is in the most perfectly designed bearing a sliding action which, however slight, must yet be provided for. If balls and raceways were absolutely incompressible there would in practice be solely point contact and, therefore, a pure rolling action, but as far as present day mechanical processes go even the most perfect of materials are somewhat elastic and there is an actual area of deformation between the ball and the raceway under load. At the extremities of this area, whose diameter is, of course, very small, sliding must exist, but the presence of a lubricant at this point renders what small amount of friction there is almost negligible. Improbable as it may seem, there is an actual film of lubricant maintained between these surfaces of contact, however minute they are, providing the lubricant be one of sufficient viscosity.

PROTECTIVE VALUE OF THE LUBRICANT

Aside from its lubricating quality, oil or grease acts as a protecting agent for the ball bearing. The finely polished surfaces of balls and raceways are subject to attack by rust through atmospheric action and may be damaged by the entrance of foreign matter from the outside, such as grit and dust. Lubricant effectually surrounding the bearing will not be penetrated by this atmospheric moisture and thus the surfaces of action are preserved in their original finely polished condition. If, however, rust or grit should be allowed to attack the carrying surfaces these are quickly destroyed and an earlier failure of the bearing may be expected.

In order that the lubricant may in itself be no source of danger, it must

respond to certain requirements; incidentally requirements exacted of the ball-bearing lubricant are thoroughly desirable in one to be used in plain bearings as well.

REQUIREMENTS OF A BALL BEARING LUBRICANT

First and most vital is the requirement that the lubricant of itself shall do no damage to the bearing, neither originally nor through deterioration. The most common fault in oils is the presence of free acid or its development. Free acid is never found in properly refined mineral or hydro-carbon oils, nor will these deteriorate due to the action of the atmosphere, and therefore the only kind of oil satisfactory for ball bearing lubrication is mineral oil. Vegetable oils, such as castor, cotton-seed, rape, linseed and the like are barred, principally because of their tendency to gum up, become rancid and develop acid. The animal oils are objectionable for the same reason. A simple test for the presence of objectionable elements in an oil or grease is to coat a finely polished surface of steel or brass or the surface of a ball bearing, with the oil or grease in question and expose it to the sunlight for a few weeks; at the end of that time the steel should show its original polish. Any discoloration or drying up of the lubricant on the steel is sufficient to bar it from consideration. The so-called petroleum greases, among which are vaseline and cosmoline, have no detrimental action in themselves, since they are derivatives of mineral oil; they have, however, low viscosity and a low melting-point—100 to 125° F.; they pound out thin in action and have little lubricating value. Their use is limited to very slow speeds and their chief advantage is low cost.

Just as acid is a thing to be guarded against in oils, so is free alkali the most common enemy to ball bearings, among the greases. The familiar yellow cup grease is usually a combination of a mineral oil and some vegetable or mineral oil or fat which latter is saponified by the addition of a caustic; the result is a lubricant having body and stiffness The saponifying material should be small in quantity and very carefully compounded, else free alkali, having a detrimental action on the steel may result The action of an alkali, as of an acid, is to pit or etch the surfaces upon which it is deposited.

The addition of mica, ground cork, wood and such substances, frequently added to overcome noise in gear cases of automobiles, is a positive menace to the ball bearing, since this foreign matter opposes free ball rotation; if it be present in large enough amount, the result may easily be that the balls are wedged between the raceways and actual fracture may result. Certainly the free rolling quality of the ball bearing will be lost.

The question of the beneficial effect of graphite in ball bearing lubrication is one often asked. The answer is simply that graphite in any shape or form that will settle and pack with time when quiescent cannot be of assistance to

the ball bearing itself. Whether it is a detriment of enough importance to offset its undoubted advantage to gears or whatever the mounting may be, depends entirely on the kind and fineness of the graphite and the proportions in which it is added to the lubricant.

It may readily be seen that any considerable amount of graphite settling in the bottom of the raceway will interfere with the free rolling of the balls in the same manner as ground cork or any other substance. The claim made for graphite, and one certainly proved in the case of the plain bearing, is that it deposits itself in a fine film on the surface upon which the shaft may revolve. A microscopic examination of the balls and grooves of a carefully made ball bearing shows almost no scratches or inequalities; after such a bearing has been run in oil and graphite of proper proportions, examination again will show that absolutely no graphite has been deposited. Frequent observations carefully made show a slight increase in the friction when graphite is present over that when oil alone is used. Such results clearly show that graphite has no value in so far as reduction of ball bearing friction may be concerned. Over the usual life of a ball bearing, however, probably no actual damage will result and considerable good may have accurred to the elements carried upon the ball bearings; such use would, of course, be justifiable. The damage, however, from graphites that settle and accumulate in the bearings cannot be overlooked.

EFFECT OF SPEED UPON CHOICE OF LUBRICANT

The effect of speed upon the choice of lubricant must likewise be considered. As a general statement, greases are desirable for use up to just as high a speed as is possible. An undue increase of temperature and a thinning out of grease will indicate the limit of its usefulness; when this is noted oil must be employed. Oil has a higher viscosity than grease and theoretically is a better lubricant, but its advantage in this regard is more than offset by the difficulty of its proper retention, in the bearing case. Such case may be solidly packed with grease that will thus not only lubricate the bearing, but will also prevent foreign substances from entry. If oil be used the ball will merely dip into the oil, splashing it over the entire bearing, but the bearing is then more open to the entry of foreign matter. There will be more internal friction in the grease than in the oil and it is this that at high speed produces excessive temperature. As a general thing greases may be used up to about 2000 to 2500 revolutions per minute.

MOUNTINGS MADE TO RETAIN LUBRICANT

In order that the lubricant may be effectually retained, various arrangements are used, depending upon the conditions surrounding the bearing. In the simplest sort of mounting for a radial bearing the shaft projects through

the casing, and the casing itself is provided with two lips, between which is a space for lubricant. If the condition be such that additional protection is needed, one additional groove is provided and this may be fitted with a cup of some sort from which grease will be steadily fed to the groove to keep that filled. This makes a definite frictionless packing. For still more severe conditions a third groove is added; in this latter groove felt is occasionally placed, whose adherence to the shaft is guaranteed by some sort of spring tension. Unfortunately, this is subject to drying out and thus loses its efficiency; when that occurs it is a positive detriment.

The single or multiple groove arrangement empty or with only grease filling is the most effective; but it is essential that the bore of the lips be not more than 1/64 in. larger than the shaft in diameter, that the lip edges be sharp instead of rounded over, and that the lips be at least 3/32 in. wide. The grooves may also be cut in the shaft, leaving bands between with sharp edges. The only objection to this is the weakness of the shaft.

HINTS ON THE CARE OF AUTOMOBILE BALL BEARINGS

The preceding paragraphs point out the need of lubrication and the necessity of keeping ball bearings free from all foreign particles. D. W Page, writing in the *American Machinist*, gives a number of hints on the care of ball bearings in automobiles. He says: "Many cases of trouble in rear axle and transmission case bearings have been traced to the presence of minute particles of metal ground off from the gears, or to particles of sand loosened from the interior of the gear case or rear axle housing castings. In sliding gears, especially when operated by an inexperienced person, there is a constant clashing of the pinions in changing speeds, which tends to loosen particles of metal from the teeth. These fall into the lubricants and ofttimes find their way into the ball races. The rapid failure of the bearing is the inevitable consequence. Likewise failures in ball bearings on automobile engine crankshafts have been traced to the presence of particles of carbon and other foreign matter in the oil.

Another source of trouble is the washing of rust and dirt into the bearings through carelessness in washing the automobile. This is particularly true of wheel bearings, where many failures have been traced directly to the presence of rust and dirt caused by the indiscriminate use when washing with a stream of water under pressure of 40 to 50 lb. per square inch; that is, the normal pressure of the city water mains.

Another practice that is detrimental to ball bearings is used in many motor car repair shops. It consists in dipping the bearings for the purpose of cleansing into dirty gasoline in which perhaps gears and other parts covered with metallic particles have already been washed.

Still another practice prevalent not only in garages and repair shops, but in too many machine shops is the employment of a hammer, particularly a hard hammer, in assembling or dismounting ball bearings. It will be remembered that these are made of hardened steel. The man who hammers a reamer or cutter is immediately discharged, but gets only a reprimand when he similarly abuses a ball bearing.

SECTION V

Roller Bearings with Flexible Rollers

Roller bearings with hollow, cylindrical, helical, flexible rollers made by the Hyatt Roller Bearing Company are used in machinery in general, on line shafting and in automobiles. The rollers are wound from flat strip steel into a closed helical coil. Thus they are flexible and can adapt themselves to slight irregularities in either journal or box without causing excessive pressure. The cylindrical hollows in rollers serve as storage space for lubricant and the helical

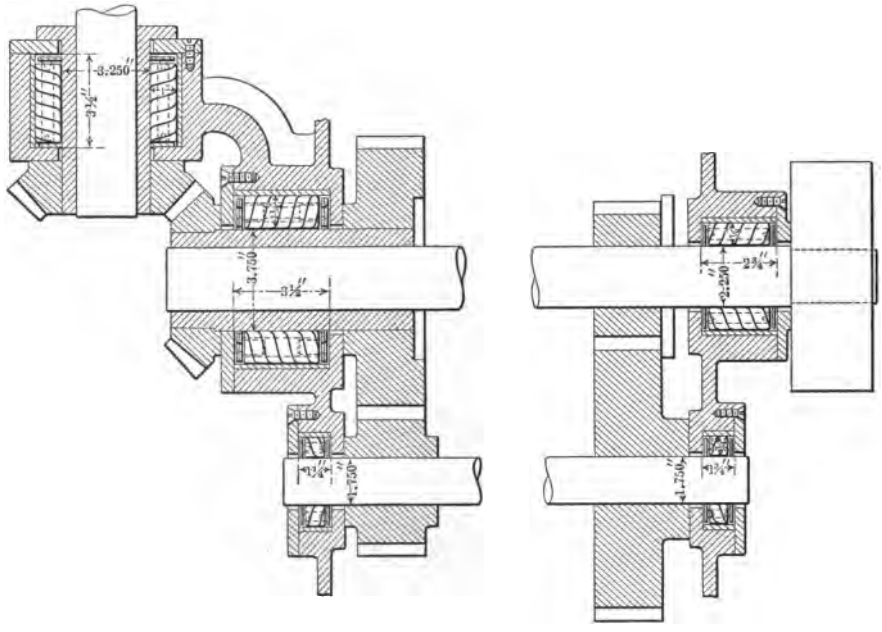


FIG. 104. FLEXIBLE ROLLER BEARING MOUNTINGS FOR TRANSMISSION SHAFTS OF MULTIPLE DRILLER.

interstices distribute the oil the entire length of the box. One-half of the rollers in a box have a right-hand helix, the other half left-hand.

In the form of bushings, two types are made, known as the standard type and the high duty type. In the standard type the rollers are of carbon steel with

an outer shell or lining of special analysis sheet steel. The rollers run in contact with the shaft or journal. Where the bearing surface is generous it is satisfactory to operate the rollers direct on soft steel surfaces.

In the second type the rollers are of nickel steel, 3 1/2 per cent. nickel, and

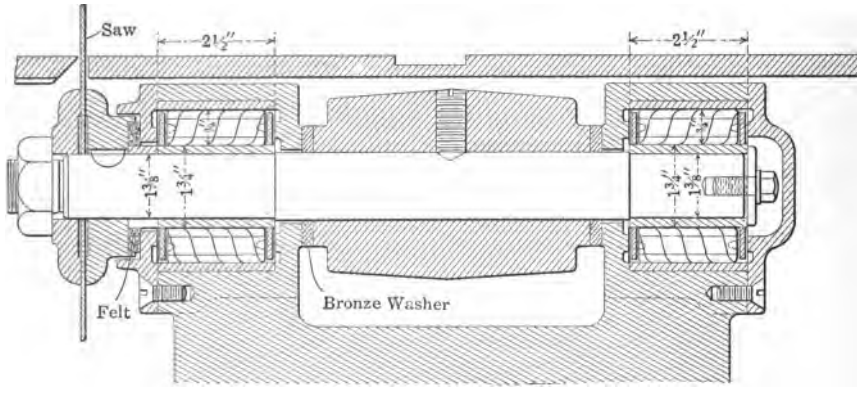


FIG. 105. FLEXIBLE ROLLER BEARINGS MOUNTING FOR A CIRCULAR SAW ARBOR.

heat treated. The lining is tubular and a tubular sleeve is provided to slip over the shaft or journal. Both of these parts are also heat treated. Several devices are used to cage the rollers, which are squared on the ends to thrust against the ends of the box. As the allowable unit bearing pressures of the high duty rollers are higher than for the standard rollers the high-duty bearings

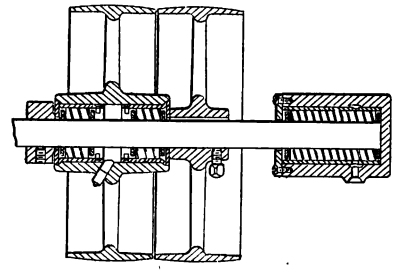


FIG. 106. FLEXIBLE ROLLER BEARING APPLIED TO COUNTERSHAFT BEARING AND LOOSE PULLEY.

have the practical advantage of being shorter for the same load than the commercial bearings.

The shafting bearing is of the standard type with a horizontally split box so that it can be put on a shaft anywhere and does not have to be slipped on over the end.

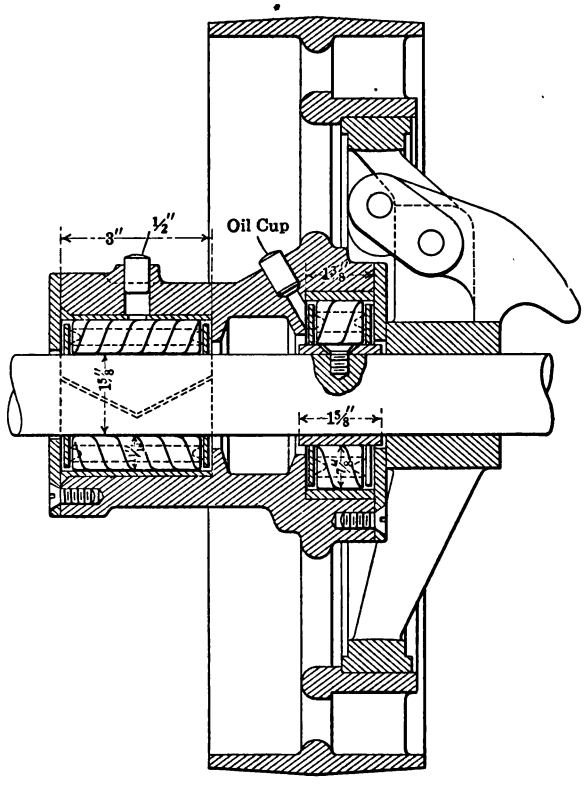


FIG. 107. FLEXIBLE ROLLER BEARING MOUNTING FOR CLUTCH PULLEY.

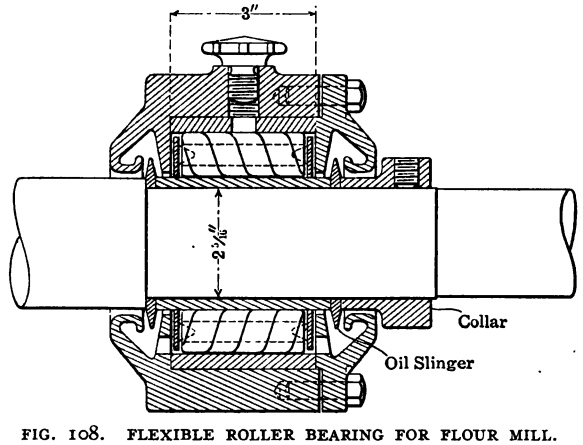


FIG. 108. FLEXIBLE ROLLER BEARING FOR FLOUR MILL.

COEFFICIENTS OF FRICTION

In the *Trans. A. S. M. E.*, Vol. XXVII, page 504, Prof. A. L Williston gives the results of various experiments with Hyatt bearings. From that source, the following coefficients of friction are taken: Average for various loads at 130 r. p. m., 0.0114; at 302 r. p. m., 0.0099; at 585 r. p. m., 0.0147. The average for very heavy loads at 185 r. p. m. was 0.011; at 215 r. p. m. 0.0093.

A test made by the author comparing Hyatt standard lineshaft bearings with babbitted bearings gave a saving for the roller bearings of 64.9 per cent. The shaft was 152 ft. long, 2 15/16 in. in diameter and was supported by 20 hangers.

ALLOWABLE PRESSURES AND SPEEDS

The allowable pressures for the standard or commercial type are fixed by the quality of the shaft or journal against which the rollers bear. For low speeds

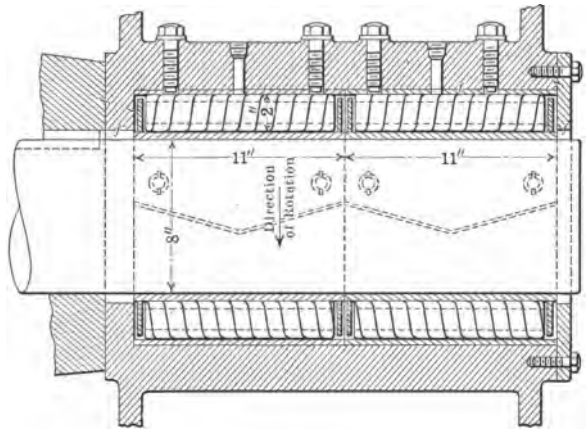


FIG. 109. FLEXIBLE ROLLER BEARING FOR PIG BREAKER.

up to 50 r. p. m. the maximum limit lies between 400 to 500 lb. per square inch of projected journal area. The method of housing, particularly in its relation to the distribution of load, and quality of lubrication have an influence in determining these limits.

With an increase of speed the allowable pressure decreases. For lineshaft bearings up to 3 15/16 in. in diameter running up to 600 r. p. m. 30 lb. per square inch is considered good practice. For larger shafts the same factor is allowable up to speeds of 400 r. p. m.

The high-duty bearings carry much greater unit loads. A rating of 750 lb. per square inch at 1000 r. p. m. is conservative.

A limiting maximum speed for the standard or commercial bearings is about 1500 r. p. m.; for the high-duty bearings, 3000 r. p. m.

LUBRICATION

These bearings must be well lubricated. If the speed is medium or high, a good body machinery oil is suitable. If the speed is slow a heavy body machinery oil or grease is better.

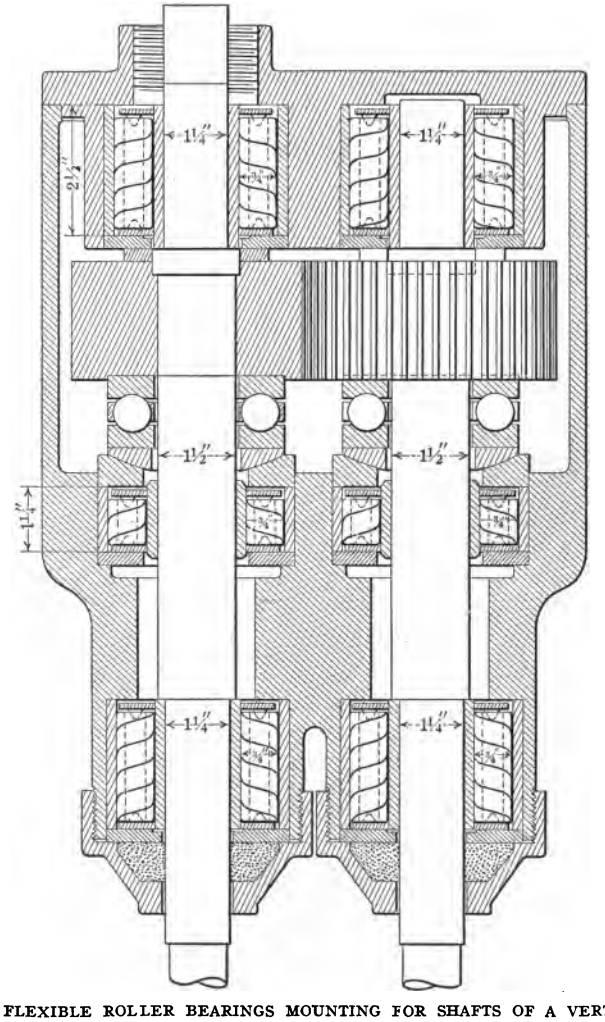


FIG. 110. FLEXIBLE ROLLER BEARINGS MOUNTING FOR SHAFTS OF A VERTICAL PUMP.

TYPICAL APPLICATIONS IN MACHINERY

Several typical designs and machinery applications are shown in Figs. 104 to 110, inclusive. The first, Fig. 104, is of shaft mountings for a multiple-spindle driller; the second, Fig. 105, is a circular saw arbor on two bearings.

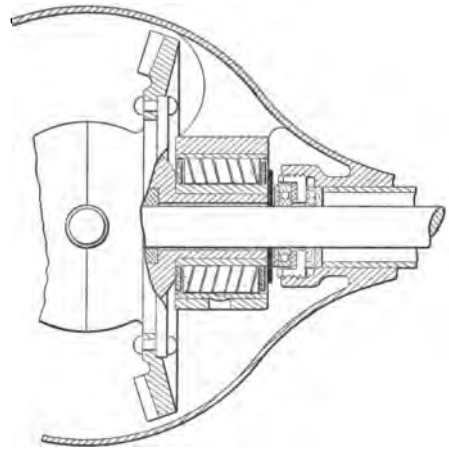


FIG. 111. STANDARD FLEXIBLE ROLLER BEARING APPLIED TO BEVEL GEAR REAR AXLE WITH BEARING ON DIFFERENTIAL HUB.

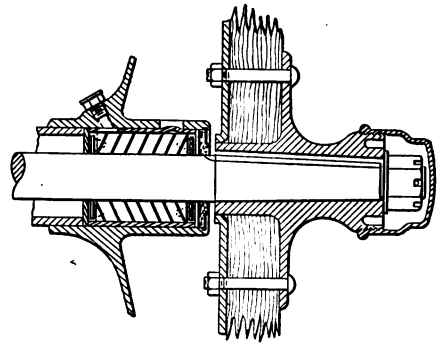


FIG. 112. STANDARD FLEXIBLE ROLLER BEARING APPLIED TO BEVEL GEAR REAR AXLE WITH BEARING MOUNTED ON SOFT AXLE.

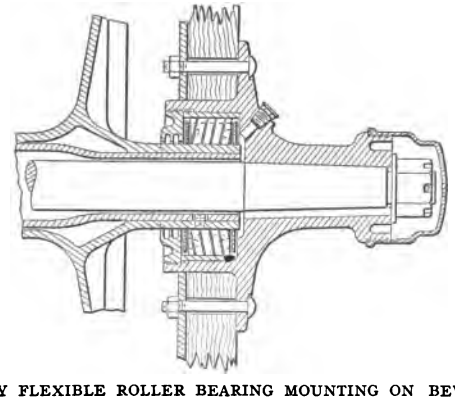


FIG. 113. HIGH DUTY FLEXIBLE ROLLER BEARING MOUNTING ON BEVEL GEAR REAR AXLE WITH BEARING ON TUBE.

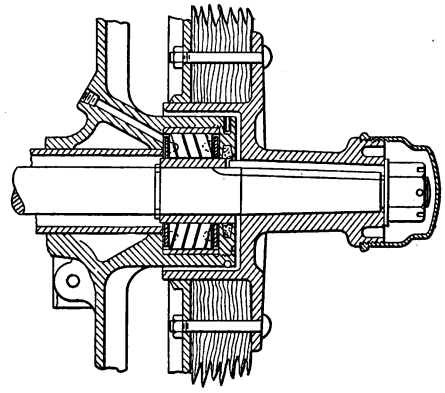


FIG. 114. HIGH DUTY FLEXIBLE ROLLER BEARING MOUNTING FOR BEVEL GEAR REAR AXLE WITH SLEEVE AND NO THRUST BEARING.

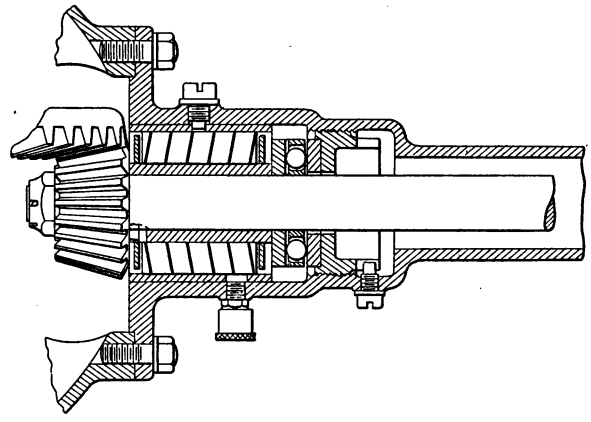


FIG. 115. HIGH DUTY FLEXIBLE ROLLER BEARING MOUNTING FOR INNER END OF BEVEL PINION SHAFT—MATING GEAR ON REAR AXLE.

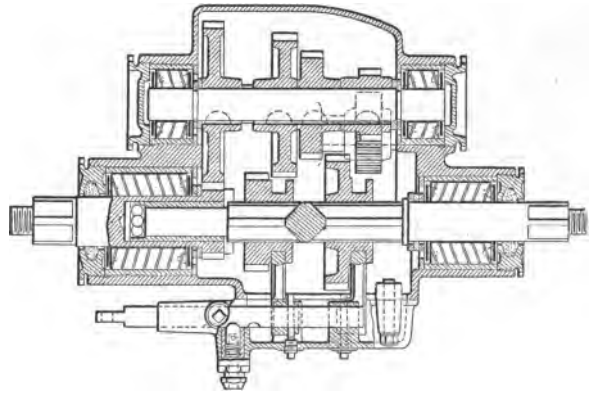


FIG. 116. HIGH-DUTY FLEXIBLE ROLLER BEARINGS IN SELECTIVE TRANSMISSION MOUNTED INDEPENDENTLY.

Figs. 106 and 107 show a countershaft bearing and loose pulley mounting, and two bearing applied to a friction clutch. In this latter example the short high-duty bearing carries the load and the longer standard bearing is a steadying bearing at the end of the hub.

Fig. 108 is a flour mill construction and is of interest as showing the device used to prevent the escape of oil.

Fig. 109 is the design of an 8-in. bearing for a pig breaker. Because of the length —22 in.—two sets of rollers are used. Fig. 110 shows the mounting of an 8-in. vertical spiral pump. Roller bearings are used for the guide bearings and ball bearings for the thrusts.

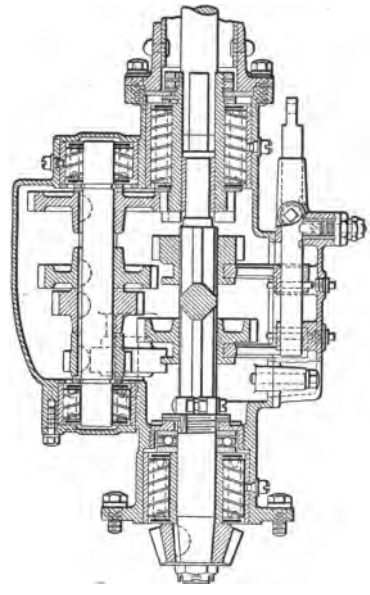


FIG. 117. HIGH-DUTY FLEXIBLE ROLLER BEARINGS IN SELECTIVE TRANSMISSION ON REAR AXLE.

IN AUTOMOBILE PRACTICE

These bearings have had an extensive use in automobiles. The seven engravings, Figs. 111 to 117 inclusive, show a few typical arrangements. Figs. 111 and 112 are of standard bearings applied to bevel gear rear axles. Figs. 113 and 114 are of high duty bearings also applied to bevel gear rear axles. Fig. 115 shows the inner bearing of a bevel pinion with a ball thrust, and Figs. 116 and 117 show two mountings for selective transmissions.

SECTION VI

Radial Roller Bearings with Solid Rollers

Solid roller journal bearings divide into two classes, the first using cylindrical rollers, and the second conical rollers. Both have their extensive applications In general, the cylindrical roller is long and small in diameter compared with its length, while the conical roller is usually short and with a lesser difference between its length and diameter. The cylindrical roller is used on machinery in general, while the conical roller has had its special extensive application in automobile practice A newer development in the use of the cylindrical roller is for the bearings of street railway car axles.

Figs. 118 and 119 show typical designs for cylindrical roller journal bearings, as made by the Standard Roller Bearing Company. The first is a

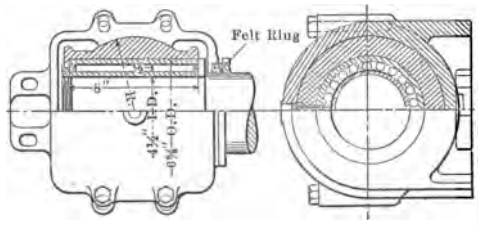


FIG. 118. ROLLER JOURNAL PEDESTAL BOX.

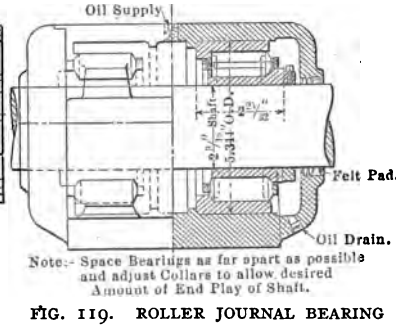


FIG. 119. ROLLER JOURNAL BEARING FOR ELECTRIC MOTOR.

design for a pedestal box. It will be noted that the rollers are comparatively small in diameter, are long, have a ball at each end to reduce the friction from thrust pressure, are held together in a cage and the bearing shell is fitted into the housing by a spherical seat to permit of self alinement. Another feature worth commenting upon is the felt rings at the end to exclude dirt and keep in lubricant.

Fig. 119 shows a motor bearing having two rows of rollers. This bearing is distinguished from the one that precedes it by the fact that the rollers are shorter, larger in diameter, and that the ends have angular faces to take up a certain amount of end thrust. These oblique faces fit between corresponding

shoulders, so that the motor or armature shaft can float endwise a short distance and be checked by the meeting of these inclined surfaces. It is evident that no great amount of end thrust can be successfully carried. The ends of the faces are provided with a felt washer and grooves to check the outward flow of oil and the entrance of dust.

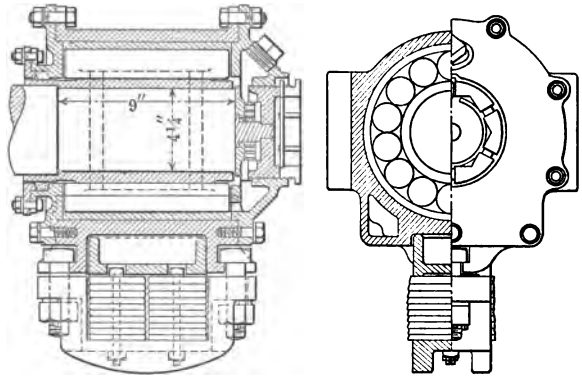


FIG. 120. ROLLER BEARING FOR ELECTRIC CAR TRUCK AXLE.

Fig. 120 shows a roller bearing designed for the axles of electric cars, this particular one being for a Halsey truck. The journal is 4 1/4 in. in diameter and 9 in. long. It is covered by a steel sleeve, upon which the rollers bear. The details of design, including provision to exclude dust and provide means for introducing lubricant and storage place for the lubricant are plainly shown.

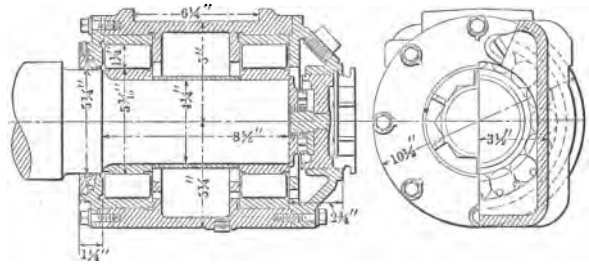


FIG. 121. ANOTHER FORM OF ROLLER BEARING FOR ELECTRIC CAR TRUCK AXLE.

At the end, in a vertical position, is a small plain roller thrust bearing to take the end thrust of the axle.

Another design of a roller bearing for this purpose is shown by Fig. 121, in which there are two sets of rollers instead of one, otherwise the general details of the bearing design are not different, except that this has a large pocket at the center for oil, with a draining plug at the bottom. This design makes

the roller bearing interchangeable with plain brass lined bearings and applicable to the standard size of thrusts in which the latter are customarily used.

Bearings of this kind have been used experimentally to quite an extent in this country and as the experimental stage can be assumed to have been passed, their adoption by railways may be looked for as a development of the next few years.

FORMULA FOR CAPACITY FOR CYLINDRICAL JOURNAL ROLLER BEARINGS

The formula for capacity for solid roller journal bearings used by the Standard Roller Bearing Company is as follows:

$$130{,}000\,\frac{d^2nl}{3s} = \text{capacity in pounds.}$$

Here d = the diameter of the rollers in inches,
n = the number of rollers,
l = the length of each roller in inches,
s = the circumferential speed of each roller in feet per minute.

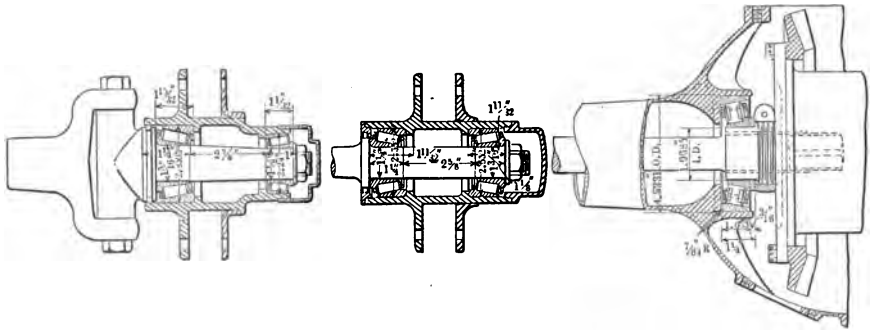


FIG. 122. CONICAL ROLLER JOURNAL BEARINGS APPLIED TO AUTOMOBILE HUBS.

FIG. 123. CONICAL ROLLER BEARING AS DIFFERENTIAL BEARING ON AN AUTOMOBILE REAR AXLE.

CONICAL ROLLER JOURNAL BEARINGS

Conical roller journal bearings have had their most extensive use in automobile practice. Figs. 122, 123 and 124 are from designs of the Standard Roller Bearing Company. Fig. 122 showing two constructions for automobile hubs having two single rows of rollers. Fig. 123 shows a bearing of this type, used as a differential bearing on a rear axle. Fig. 124 shows in section a double taper roller bearing.

FORMULA FOR CAPACITIES FOR TAPERED OR CONICAL JOURNAL BEARINGS

The load carrying capacity in pounds is given by the following formula, which is from the practice of the Standard Roller Bearing Company:

$$130{,}000\frac{d^2nl}{3s} = \text{capacity in pounds.}$$

In which d = the mid-diameter of the rollers in inches,
n = the number of rollers,
l = the contact length of a roller with its bearing washer in inches,
s = the mean circumferential speed of each roller in feet per minute.

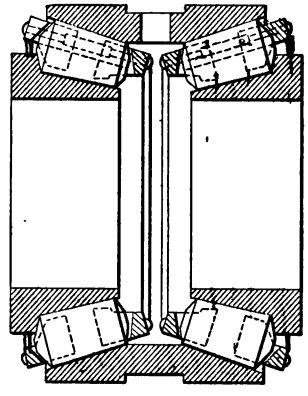


FIG. 124. DOUBLE CONICAL JOURNAL BEARING.

SECTION VII

Roller Thrust Bearings

The Standard Roller Bearing Company is the builder of a patented type of plain roller thrust bearings that has been adapted to a wide field including situations where very heavy loads are to be sustained. It has had an extensive application for water turbine generators; in fact, its first use for heavy loads was in connection with the turbines at Niagara.

The design consists essentially of two flat plates or washers, between which is a bronze cage containing a number of cylindrical rollers. These rollers are

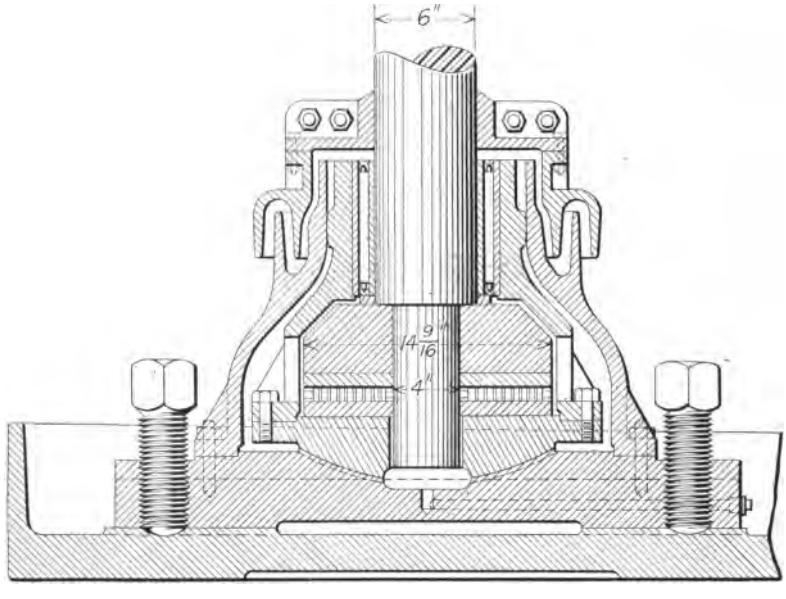


FIG. 125. PLAIN ROLLER STEP BEARING APPLIED TO A CHEMICAL FURNACE.

comparatively short for their diameters, have spherical ends and are fitted into radial slots in the cage. The outermost roller of each row is recessed to take a ball, which is held in a hole in a steel band surrounding the cage. A second band then surrounds the first and keeps the balls in position. These balls are used to reduce the friction due to end thrust from centrifugal force. The washers and cages are made either split or solid according to conditions.

It is evident that the motion of the roller is a combination of rolling and sliding, for cylindrical rollers are traveling in a circular ~~pitch~~.

These bearings are made for all kinds of load and service, from a small bearing adapted to a 7/8 in. shaft up to a single bearing supporting a load of over 2,000,000 lb.

Figs. 125, 126 and 127 show three of these bearings. The first is for a 6 in. shaft and was designed for a revolving chemical furnace. Figs. 126 and 127 are of larger bearings, the latter being for the turbo-generators at McCall's

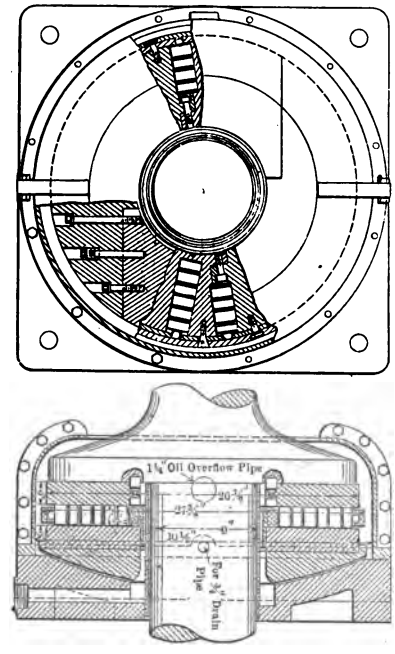


FIG. 126. PLAIN ROLLER THRUST BEARING FOR NINE-INCH SHAFT, WITH SHAFT PASSING THROUGH BEARING PROPER.

Ferry on the Susquehanna River. Five bearings of this design have been installed there on the main turbine generators, three sustaining a load each of 330,000 lb., and two a load each of 410,000 lb. The size of the bearing can be realized when the shaft diameter is seen to be 21 in.

The thrust is transmitted to the bearing through a collar 45 1/2 in. in diameter forged integral with the shaft.

In every bearing two kinds of rollers are used—short and long. These are so arranged that they do not track, but cover the working surfaces of the washers so that the latter are stressed uniformly. To insure perfect alinement of the

two washers it is customary to make the bottom of the lower one hemispherical in shape and seated correspondingly or to use a pair of leveling plates.

Lubrication is an important point. In the larger bearings flooded lubrication is maintained. The smaller sixes run in a bath of oil.

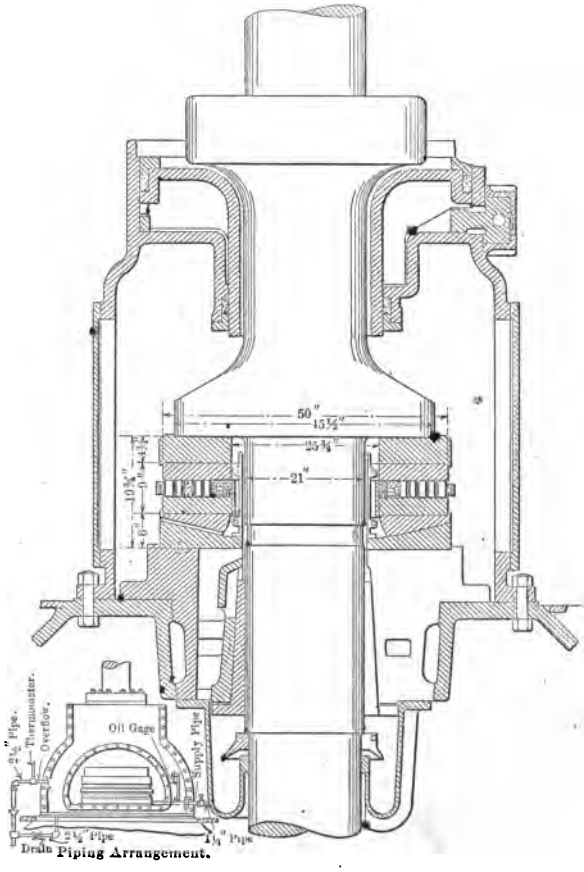


FIG. 127. PLAIN ROLLER THRUST BEARING—TYPE INSTALLED AT MCCALL'S FERRY.

FORMULA FOR CAPACITY

The formula for capacity of these bearings, based finally on a careful analysis of a large number of bearings in successful use, is dependent upon the roller diameter, the mean circumferential speed of the roller and the sum of their lengths. This formula is:

$$100{,}000\frac{d^2M}{S} = \text{load capacity in pounds.}$$

In which D = the diameter of the rollers in inches;
M = the sum of the lengths of all the rollers in inches;
S = the mean circumferential speed of the rollers in feet per minute.

The value 100,000 of the constant factor is a minimum and thereby gives a minimum value for the capacity. Owing to advantageous conditions, this numerical factor may be considerably larger in some designs.

Experience has shown that good practice sets a maximum limit for roller speeds of about 3000 r. p. m. Similarly the maximum circumferential speed of the roller is fixed at about 3000 ft. per minute. The factors of a uniform or variable speed and a uniform or variable load, enter to modify the given numerical constant.

TEST OF BEARINGS AT McCALL'S FERRY

An extensive series of tests has been run on the bearings at McCall's Ferry to determine the friction loss. Valuable data have been obtained.

A word of explanation in regard to this installation is necessary. At the time of testing five units were in operation, three of 7500 and two of 10,000 kilovolt amperes rating. Each turbine has a rating of 13,500 h. p., with a head of 53 ft. and consists of two Francis wheels, located in tandem on the same shaft, the lower one being below the tail race level. The generators are three-phase, 25-cycle, 11,000 volts at a speed of 94 r. p. m. The first unit at the time of this writing has been in operation for over one year.

Each bearing is lubricated by flood lubrication, circulation being maintained by gravity with the supply taken from an elevated tank. The overflow from each bearing is to a filter, from whence it is pumped into the storage tank. The filters are provided with cooling coils for circulating water, but these are seldom used except in the very hottest summer weather. Oil enters the bearing through the lower leveling washer into the space outside of the vertical oil guard. Thence it passes upward and by means of centrifugal action of the rollers passes outward through the openings in the cage and between the rollers themselves, and overflows at a level about midway of the height of the upper bearing washer.

In case the oil supply should be cut off for any reason whatever, circulation will still be automatically maintained by the pumping action of the rollers and through oil grooves in the lower leveling washer.

Each turbo-generator has three babbitted guide bearings and two lignum vitæ guide bearings, in addition to the roller thrust.

The method of test was to measure accurately the rise in temperature of the oil as it passed out of the bearing, to determine carefully the amount of oil in the bearing, to find the unit heat capacity of the oil, and to determine the heat loss

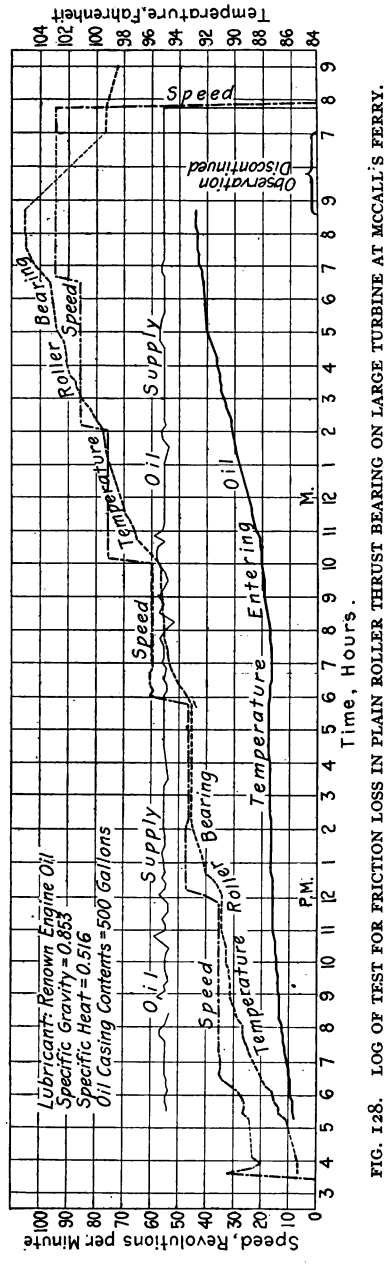


FIG. 128. LOG OF TEST FOR FRICTION LOSS IN PLAIN ROLLER THRUST BEARING ON LARGE TURBINE AT MCCALL'S FERRY.

from the bearing per unit of time, due to radiation, dissipation and convection of heat. A proper handling of these data gave the amount of heat liberated in each bearing during each test. Thus it involved the friction of the rollers rolling and sliding on their bearing plates, the friction of the rollers between themselves and between the band; in fact, all of the frictional losses in the thrust bearing itself. Any losses taking place in any other bearing in the machine in no wise affected the observations or results.

Fig. 128 shows the graphic log of a test of one of the more heavily loaded bearings. It will be seen that the speed was varied in six steps, that the temperature of the entering oil and its quantity were kept practically uniform. The temperature of the issuing oil is seen to rise with the various steps of the increas-

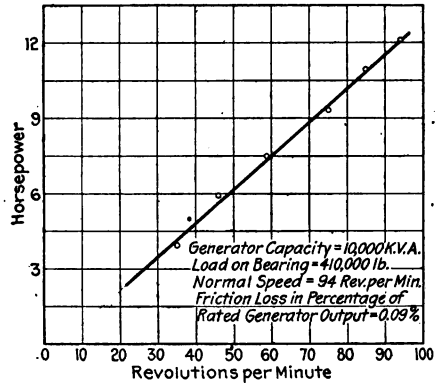


FIG. 129. HORSEPOWER LOSS AT VARIOUS SPEEDS IN A LARGE PLAIN ROLLER BEARING. SEE FIG. 128.

ing speed. At the extreme right is the temperature drop per unit of time after the turbine was shut down. At the left hand end are data giving the specific gravity and specific heat of the lubricating oil.

Fig. 129 shows the horsepower loss curve, corresponding with the test, graphically shown in Fig. 128. It will be noticed that at normal speed the total loss is slightly greater than 12 h. p. Similarly Fig. 130 shows the power loss curve for the same kind and size of bearing, but with a load of 330,000 lb., or 80,000 lb. less than for Fig. 129. Here will be noticed that the horsepower loss at normal speed is slightly greater than 10 h. p.

Fig. 131 similarly shows a horsepower loss curve for a much smaller bearing, a bearing used on an exciter. Here the load is 22,300 pounds.

A consideration of these curves shows that in every case the curve is a straight line, indicating that the loss is directly proportional to the speed. Further a comparison of Figs. 129 and 130 shows that the loss is roughly proportional to the load.

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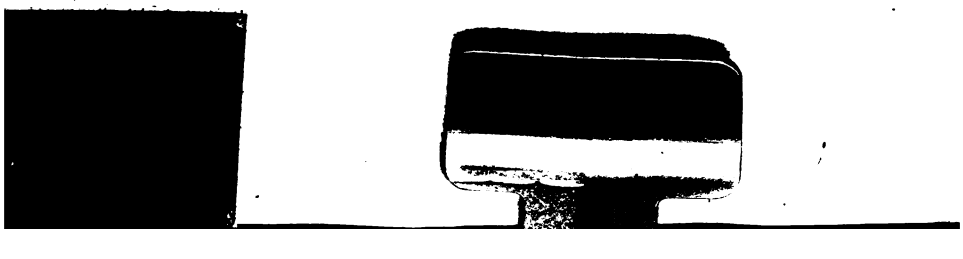
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